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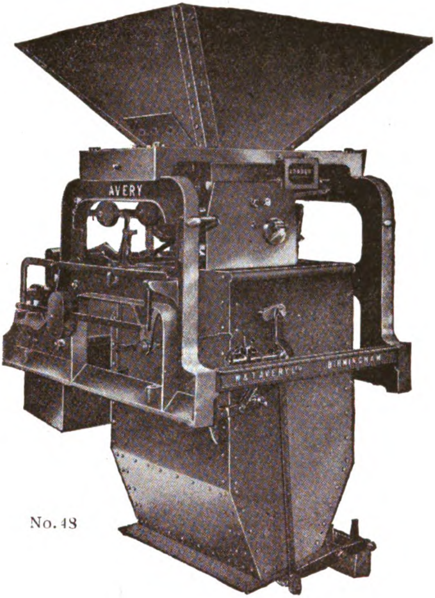

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Entirely Re-written

796 pages
With 256 Illustrations and 175 Tables

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PREFACE
TO THE
THIRD EDITION

In recasting the general scheme of this book it has been considered desirable *inter alia* to deal with the various types of prime movers in a very complete manner, and it is hoped that the treatment of their respective subjects by the several specialists who have written Section VI will be regarded as attaining fully this object. The rest of the work covers generally mechanical engineering details, except that in Section I are mathematical tables and data of use to engineers.

The scope of the field covered is such that the book should be useful to all holders of senior positions such as Consulting Engineers, Designers, Draughtsmen, Works Managers, also to superintendents or engineers-in-charge of existing plant, and also to students and junior engineers.

Our very best thanks are due to the various firms who kindly supplied drawings and data.

W. ERSKINE DOMMETT.

SURBITON, 1922.

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SECTION I

MATHEMATICAL TABLES

TABLE I.

SQUARES AND CUBES OF NUMBERS.

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
⅛	0·015625	0·001935	6	36·0000	216·0000
¼	0·062500	0·015625	6¼	39·0625	244·1406
⅜	0·140625	0·052734	6½	42·2500	274·6250
½	0·250000	0·125000	6¾	45·5625	307·5468
⅝	0·390625	0·244140			
¾	0·562500	0·421875	7	49·0000	343·0000
⅞	0·765625	0·669921	7¼	52·5625	381·0781
			7½	56·2500	421·8750
1	1·000000	1·000000	7¾	60·0625	465·4843
1⅛	1·265626	1·423828			
1¼	1·562500	1·953125	8	64·0000	512·0000
1⅜	1·890625	2·599609	8¼	68·0625	561·5156
1½	2·250000	3·375000	8½	72·2500	614·1250
1⅝	2·640625	4·291015	8¾	76·5625	669·9218
1¾	3·062500	5·359375			
1⅞	3·515625	6·591796	9	81·0000	729·0000
			9¼	85·5625	791·4531
2	4·000000	8·000000	9½	90·2500	857·3750
2⅛	4·515625	9·595703	9¾	95·0625	926·8593
2¼	5·062500	11·39062			
2⅜	5·640625	13·39648	10	100·000	1000·000
2½	6·250000	15·62500	10¼	105·062	1076·890
2⅝	6·890625	18·08789	10½	110·250	1157·625
2¾	7·562500	20·79687	10¾	113·562	1242·296
2⅞	8·265625	23·76367			
			11	121·000	1331·000
3	9·000000	27·00000	11¼	126·562	1423·828
3¼	10·56250	34·32812	11½	132·250	1520·875
3½	12·25000	42·87500	11¾	138·062	1622·234
3¾	14·06250	52·73437			
			12	144·000	1728·000
4	16·00000	64·00000	12¼	150·062	1838·265
4¼	18·06250	76·76562	12½	156·250	1953·125
4½	20·2500	91·12500	12¾	162·562	2072·671
4¾	22·5625	107·1718			
			13	169·000	2197·000
5	25·0000	125·0000	13¼	175·562	2326·203
5¼	27·5625	144·7031	13½	182·250	2460·375
5½	30·2500	166·3750	13¾	189·062	2599·609
5¾	33·0625	190·1093			

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
14	196·000	2744·000	25	625·00	15625·000
14¼	203·062	2893·640	25½	650·25	16581·375
14½	210·250	3048·625	26	676·00	17576·000
14¾	217·562	3209·046	26½	702·25	18609·625
			27	729·00	19683·000
15	225·000	3375·000	27½	756·25	20796·875
15¼	232·562	3546·578	28	784·00	21952·000
15½	240·250	3723·875	28½	812·25	23149·125
15¾	248·062	3906·984	29	841·00	24389·000
			29½	870·25	25672·375
16	256·000	4096·000			
16¼	264·062	4291·015	30	900·00	27000·000
16½	272·250	4492·125	30½	930·25	28372·625
16¾	280·562	4699·421	31	961·00	29791·000
			31½	992·25	31255·875
17	289·000	4913·000	32	1024·00	32768·000
17¼	297·562	5132·953	32½	1056·25	34328·125
17½	306·250	5359·375	33	1089·00	35937·000
17¾	315·062	5592·359	33½	1122·25	37595·375
			34	1156·00	39304·000
18	324·000	5832·000	34½	1190·25	41063·625
18¼	333·062	6078·391	35	1225·00	42875·000
18½	342·250	6363·625	35½	1260·25	44738·875
18¾	351·562	6591·796	36	1296·00	46656·000
			36½	1332·25	48627·125
19	361·000	6859·000	37	1369·00	50653·000
19¼	370·562	7133·328	37½	1406·25	52734·375
19½	380·250	7414·875	38	1444·00	54872·000
19¾	390·062	7703·734	38½	1482·25	57066·625
			39	1521·00	59319·000
20	400·00	8000·000	39½	1560·25	61629·875
20½	420·25	8615·125			
21	441·00	9261·000	40	1600·00	64000·000
21½	462·25	9938·375	40½	1640·25	66430·125
22	484·00	10648·000	41	1681·00	68921·000
22½	506·25	11390·625	41½	1722·25	71473·375
23	529·00	12167·000	42	1764·00	74088·000
23½	552·25	12977·875	42½	1806·25	76765·625
24	576·00	13824·000	43	1849·00	79507·000
24½	600·25	14706·125	43½	1892·25	82312·875

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
44	1936·00	85184·000	63	3969·00	250047·000
44½	1980·25	88121·125	63½	4032·25	256047·875
45	2025·00	91125·000	64	4096·00	262144·000
45½	2070·25	94196·375	64½	4160·25	268336·125
46	2116·00	97336·000	65	4225·00	274625·000
46½	2162·25	100554·625	65½	4290·25	281011·375
47	2209·00	103823·000	66	4356·00	287496·000
47½	2256·25	107171·875	66½	4422·25	294079·625
48	2304·00	110592·000	67	4489·00	300763·000
48½	2352·25	114084·125	67½	4556·25	307546·875
49	2401·00	117649·000	68	4624·00	314432·000
49½	2450·25	121287·375	68½	4692·25	321419·125
			69	4761·00	328509·000
50	2500·00	125000·000	69½	4830·25	335702·375
50½	2550·25	128787·625			
51	2601·00	132651·000	70	4900·00	343000·000
51½	2652·25	136590·875	70½	4970·25	350402·625
52	2704·00	140608·000	71	5041·00	357911·000
52½	2756·25	144703·125	71½	5112·25	365525·875
53	2809·00	148877·000	72	5184·00	373248·000
53½	2862·25	153130·375	72½	5256·25	381078·125
54	2916·00	157464·000	73	5329·00	389017·000
54½	2970·25	161878·625	73½	5402·25	397065·375
55	3025·00	166375·000	74	5476·00	405224·000
55½	3080·25	170953·875	74½	5550·25	413493·625
56	3136·00	175616·000	75	5625·00	421875·000
56½	3192·25	180362·125	75½	5700·25	430368·875
57	3249·00	185193·000	76	5776·00	438976·000
57½	3306·25	190109·375	76½	5852·25	447697·125
58	3364·00	195112·000	77	5929·00	456533·000
58½	3422·25	200201·625	77½	6006·25	465484·375
59	3481·00	205379·000	78	6084·00	474552·000
59½	3540·25	210644·875	78½	6162·25	483736·625
			79	6241·00	493039·000
60	3600·00	216000·000	79½	6330·25	503254·875
60½	3660·25	221445·125			
61	3721·00	226981·000	80	6400·00	512000·000
61½	3782·25	232608·375	80½	6480·25	521660·125
62	3844·00	238328·000	81	6561·00	531441·000
62½	3906·25	244140·625	81½	6642·25	541343·375

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
82	6724·00	551368·000	102	10404	1061208
82½	6806·25	561515·625	103	10609	1092727
83	6889·00	571787·000	104	10816	1124864
83½	6972·25	582128·875	105	11025	1157625
84	7056·00	592704·000	106	11236	1191016
84½	7140·25	603351·125	107	11449	1225043
85	7205·00	612425·000	108	11664	1259712
85½	7310·25	625026·375	109	11881	1295029
86	7396·00	636056·000			
86½	7482·25	647214·625	110	12100	1331000
87	7569·00	658503·000	111	12321	1367631
87½	7656·25	669921·875	112	12544	1404928
88	7744·00	681472·000	113	12769	1442897
88½	7832·25	693154·125	114	12996	1481544
89	7921·00	704969·000	115	13225	1520875
89½	8010·25	716917·375	116	13456	1560896
			117	13689	1601613
90	8100·00	729000·000	118	13924	1643032
90½	8190·25	741217·625	119	14161	1685159
91	8281·00	753571·000			
91½	8372·25	766060·875	120	14400	1728000
92	8464·00	778688·000	121	14641	1771561
92½	8556·25	791453·125	122	14884	1815848
93	8649·00	804357·000	123	15129	1860867
93½	8742·25	817400·375	124	15376	1906624
94	8836·00	830584·000	125	15625	1953125
94½	8930·25	843908·625	126	15876	2000376
95	9025·00	857375·000	127	16129	2048383
95½	9120·25	870983·875	128	16384	2097152
96	9216·00	884736·000	129	16641	2146689
96½	9322·25	899597·125			
97	9409·00	912673·000	130	16900	2197000
97½	9506·25	926859·375	131	17161	2248091
98	9604·00	941192·000	132	17424	2299968
98½	9702·25	955671·625	133	17689	2352637
99	9801·00	970299·000	134	17956	2406104
99½	9900·25	985074·875	135	18225	2460375
			136	18496	2515456
100	10000	1000000	137	18769	2571353
101	10201	1030301	138	19044	2628072

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
139	19321	2685619	175	30625	5359375
			176	30976	5451776
140	19600	2744000	177	31329	5545233
141	19881	2803221	178	31684	5639752
142	20164	2863288	179	32041	5735339
143	20449	2924207			
144	20736	2985984	180	32400	5832000
145	21025	3048625	181	32761	5929741
146	21316	3112136	182	33124	6028568
147	21609	3176523	183	33489	6128487
148	21904	3241792	184	33856	6229504
149	22201	3307949	185	34225	6331625
			186	34596	6434856
150	22500	3375000	187	34969	6539203
151	22801	3442951	188	35344	6644672
152	23104	3511808	189	35721	6751269
153	23409	3581577			
154	23716	3652264	190	36100	6859000
155	24025	3723875	191	36481	6967871
156	24336	3796416	192	36864	7077888
157	24649	3869893	193	37249	7189517
158	24964	3944312	194	37636	7301384
159	25281	4019679	195	38025	7414875
			196	38416	7529536
160	25600	4096000	197	38809	7645373
161	25921	4173281	198	39204	7762392
162	26244	4251528	199	39601	7880599
163	26569	4330747			
164	26896	4410944	200	40000	8000000
165	27225	4492125	201	40401	8120601
166	27556	4574296	202	40804	8242408
167	27889	4657463	203	41209	8365427
168	28224	4741632	204	41616	8489664
169	28561	4826809	205	42025	8615125
			206	42436	8741816
170	28900	4913000	207	42849	8869743
171	29241	5000211	208	43264	8998912
172	29584	5088448	209	43681	9129329
173	29929	5177717			
174	30276	5268024	210	44100	9261000

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube N^3
211	44521	9393931	248	61504	15252992
212	44944	9528128	249	62001	15438249
213	45369	9663597			
214	45796	9800344	250	62500	15625000
215	46225	9938375	251	63001	15813251
216	46656	10077696	252	63504	16003008
217	47089	10218313	253	64009	16194277
218	47524	10360232	254	64516	16387064
219	47961	10503459	255	65025	16581375
			256	65536	16777216
220	48400	10648000	257	66049	16974593
221	48841	10793861	258	66564	17173512
222	49284	10941048	259	67081	17373979
223	49729	11089567			
224	50176	11239424	260	67600	17576000
225	50625	11390625	261	68121	17779581
226	51076	11543176	262	68644	17984728
227	51529	11697083	263	69169	18191447
228	51984	11852352	264	69696	18399744
229	52441	12008989	265	70225	18609625
			266	70756	18821096
230	52900	12167000	267	71289	19034163
231	53361	12326391	268	71824	19248832
232	53824	12487168	269	72361	19465109
233	54289	12649337			
234	54756	12812904	270	72900	19683000
235	55225	12977875	271	73441	19902511
236	55696	13144256	272	73984	20123648
237	56169	13312053	273	74529	20346417
238	56644	13481272	274	75076	20570824
239	57121	13651919	275	75625	20796875
			276	76176	21024576
240	57600	13824000	277	76729	21253933
241	58081	13997521	278	77284	21484952
242	58564	14172488	279	77841	21717639
243	59049	14348907			
244	59536	14526784	280	78400	21952000
245	60025	14706125	281	78961	22188041
246	60516	14886936	282	79524	22425768
247	61009	15069223	283	80089	22665187

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
284	80656	22906304	320	102400	32768000
285	81225	23149125	321	103041	33076161
286	81796	23393656	322	103684	33386248
287	82369	23639903	323	104329	33698267
288	82944	23887872	324	104976	34012224
289	83521	24137569	325	105625	34328125
			326	106276	34645976
290	84100	24389000	327	106929	34965783
291	84681	24642171	328	107584	35287552
292	85624	24897088	329	108241	35611289
293	85849	25153757			
294	86436	25412184	330	108900	35937000
295	87025	25672375	331	109561	36264691
296	87616	25934836	332	110224	36594368
297	88209	26198073	333	110889	36926037
298	88804	26463592	334	111556	37259704
299	89401	26730899	335	112225	37595375
			336	112896	37933056
300	90000	27000000	337	113569	38272753
301	90601	27270901	338	114244	38614472
302	91204	27543608	339	114921	38958219
303	91809	27818127			
304	92416	28094464	340	115600	39304000
305	93025	28372625	341	116281	39651821
306	93636	28652616	342	116964	40001688
307	94249	28934443	343	117649	40353607
308	94864	29218112	344	118336	40707584
309	95481	29503609	345	119025	41063625
			346	119716	41421736
310	96100	29791000	347	120409	41781923
311	96721	30080231	348	121104	42144192
312	97344	30371328	349	121801	42508549
313	97969	30664297			
314	98596	30959144	350	122500	42875000
315	99225	31255875	351	123201	43243551
316	99856	31554496	352	123904	43614208
317	100489	31855013	353	124609	43986977
318	101124	32157432	354	125316	44361864
319	101761	32461759	355	126025	44738875
			356	126736	45118016

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
357	127449	45499293	393	154449	60698457
358	128164	45882712	394	155236	61162984
359	128881	46268279	395	156025	61629875
			396	156816	62099136
360	129600	46656000	397	157609	62570773
361	130321	47045881	398	158404	63044792
362	131044	47437928	399	159201	63521199
363	131769	47832147			
364	132496	48228544	400	160000	64000000
365	133225	48627125	401	160801	64481201
366	133956	49027896	402	161604	64964808
367	134689	49430863	403	162409	65450827
368	135424	49836032	404	163216	65939264
369	136161	50243409	405	164025	66430125
			406	164836	66923416
370	136900	50653000	407	165649	67419143
371	137641	51064811	408	166464	67911312
372	138384	51478848	409	167281	68417929
373	139129	51895117			
374	139876	52313624	410	168100	68921000
375	140625	52734375	411	168921	69426531
376	141376	53157376	412	169744	69934528
377	142129	53582633	413	170569	70444997
378	142884	54010152	414	171396	70957944
379	143641	54439939	415	172225	71473375
			416	173056	71991296
380	144400	54872000	417	173889	72511713
381	145161	55306341	418	174724	73034632
382	145924	55742968	419	175561	73560059
383	146689	56181887			
384	147456	56623104	420	176400	74088000
385	148225	57066625	421	177241	74618461
386	148996	57512456	422	178084	75151448
387	149769	57960603	423	178929	75686967
388	150544	58411072	424	179776	76225024
389	151321	58863869	425	180625	76765625
			426	181476	77308776
390	152100	59319000	427	182329	77854483
391	152881	59776471	428	183184	78402752
392	153664	60236288	429	184041	78953589

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
430	184900	79507000	467	218089	101847563
431	185761	80062991	468	219024	102503232
432	186624	80621568	469	219961	103161709
433	187489	81182737			
434	188356	81746504	470	220900	103823000
435	189225	82312875	471	221841	104487111
436	190096	82881856	472	222784	105154048
437	190969	83453453	473	223729	105828817
438	191844	84027672	474	224676	106496424
439	192721	84604519	475	225625	107171875
			476	226576	107850176
440	193600	85184000	477	227529	108531333
441	194481	85766121	478	228484	109215352
442	195364	86350388	479	229441	109902239
443	196249	86938307			
444	197136	87528384	480	230400	110592000
445	198025	88121125	481	231361	111284641
446	198916	88716536	482	232324	111980168
447	199809	89314623	483	233289	112678587
448	200704	89915392	484	234256	113379904
449	201601	90518849	485	235225	114084125
			486	236196	114791256
450	202500	91125000	487	237169	115501303
451	203401	91733851	488	238144	116214272
452	204304	92345408	489	239121	116930169
453	205209	92959677			
454	206106	93576664	490	240100	117649000
455	207025	94196375	491	241081	118370771
456	207936	94818816	492	242064	119095488
457	208849	95443993	493	243049	119823157
458	209764	96071912	494	244036	120553784
459	210681	96702579	495	245025	121287375
			496	246016	122023936
460	211600	97336000	497	247009	122763473
461	212521	97972181	498	248004	123505992
462	213444	98611128	499	249001	124251499
463	214369	99252847			
464	215296	99897345	500	250000	125000000
465	216225	100544625	501	251001	125751501
466	217156	101194696	502	252004	126506008

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
503	253009	127263527	540	291600	157464000
504	254016	128024064	541	292681	158340421
505	255025	128787625	542	293764	159220088
506	256036	129554216	543	294849	160103007
507	257049	130323843	544	295936	160989184
508	258064	131096512	545	297025	161878625
509	259081	131872229	546	298116	162771336
			547	299209	163667323
510	260100	132651000	548	300304	164566592
511	261121	133432831	549	301401	165469149
512	262144	134217728			
513	263169	135005697	550	302500	166375000
514	264196	135796744	551	303601	167284151
515	265225	136590875	552	304704	168196608
516	266256	137388096	553	305809	169112377
517	267289	138188413	554	306916	170031464
518	268324	138991832	555	308025	170953875
519	269361	139798359	556	309136	171879616
			557	310249	172808693
520	270400	140608000	558	311364	173741112
521	271441	141420761	559	312481	174676879
522	272484	142236648			
523	273529	143055667	560	313600	175616000
524	274576	143877824	561	314721	176558481
525	275625	144703125	562	315844	177504328
526	276676	145531576	563	316969	178453547
527	277729	146363183	564	318096	179406144
528	278784	147197952	565	319225	180362125
529	279841	148035889	566	320356	181321496
			567	321489	182284263
530	280900	148877000	568	322624	183250432
531	281961	149721291	569	323761	184220009
532	283024	150568768			
533	284089	151419437	570	324900	185193000
534	285156	152273304	571	326041	186169411
535	286225	153130375	572	327184	187149248
536	287296	153990656	573	328329	188132517
537	288369	154854153	574	329476	189119224
538	289444	155720872	575	330625	190109375
539	290521	156590819	576	331776	191102676

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
577	332929	192100033	613	375769	230346397
578	334084	193100552	614	376996	231475544
579	335241	194104539	615	378225	232608375
			616	379456	233744896
580	336400	195112000	617	380689	234885113
581	337561	196122941	618	381924	236029032
582	338724	197137368	619	383161	237176659
583	339889	198155287			
584	341056	199176704	620	384400	238328000
585	342225	200201625	621	385641	239483061
586	343396	201230456	622	386884	240641848
587	344569	202262003	623	388129	241804367
588	345744	203297472	624	389376	242970624
589	346921	204336469	625	390625	244140625
			626	391876	245314376
590	348100	205379000	627	393129	246491883
591	349281	206425071	628	394384	247673152
592	350464	207474688	629	395641	248858189
593	351649	208527857			
594	352836	209584584	630	396900	250047000
595	354025	210644875	631	398161	251239591
596	355216	211708736	632	399424	252435968
597	356409	212776173	633	400689	253636137
598	357604	213847192	634	401956	254840104
599	358801	214921799	635	403225	256047875
			636	404496	257259456
600	360000	216000000	637	405769	258474853
601	361201	217081801	638	407044	259694072
602	362404	218167208	639	408321	260917119
603	363609	219256227			
604	364816	220348864	640	409600	262144000
605	366025	221455125	641	410881	263374721
606	367236	222545016	642	412164	264609288
607	368449	223648543	643	413449	265847707
608	369664	224755712	644	414736	267089984
609	370881	225866529	645	416025	268836125
			646	417316	269585136
610	372100	226981000	647	418609	270840023
611	373321	228099131	648	419904	272097792
612	374544	229220928	649	421201	273359449

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
650	422500	274625000	687	471969	324242703
651	423801	275894451	688	473344	325660672
652	425104	277167808	689	474721	327082769
653	426409	278445077			
654	427716	279726264	690	476100	328509000
655	429025	281011375	691	477481	329939371
656	430336	282300416	692	478864	331373888
657	431649	283593393	693	480249	332812557
658	432964	284890312	694	481636	334255384
659	434281	286191179	695	483025	335702375
			696	484416	337153536
660	435600	287496000	697	485809	338608873
661	436921	288804781	698	487204	340068392
662	438244	290117528	699	488601	341532099
663	439569	291434247			
664	440896	292754944	700	490000	343000000
665	442225	294079625	701	491401	344472101
666	443556	295408296	702	492804	345948088
667	444889	296740963	703	494209	347428927
668	446224	298077632	704	495616	348913664
669	447561	299418309	705	497025	350402625
			706	498436	351895816
670	448900	300763000	707	499849	353393243
671	450241	302111711	708	501264	354894912
672	451584	303464448	709	502681	356400829
673	452929	304821217			
674	454276	306182024	710	504100	357911000
675	455625	307546875	711	505521	359425431
676	456976	308915776	712	506944	360944128
677	458329	310288733	713	508369	362467097
678	459684	311665752	714	509796	363994344
679	461041	313046839	715	511225	365525875
			716	512656	367061696
680	462400	314432000	717	514089	368601813
681	463761	315821241	718	515524	370146232
682	465124	317214568	719	516961	371694959
683	466489	318611987			
684	467856	320013504	720	518400	373248000
685	469225	321419125	721	519841	374805361
686	470596	322828856	722	521284	376367048

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
723	522729	377933067	760	577600	438976000
724	524176	379503424	761	579121	440711081
725	525625	381078125	762	580644	442450728
726	527076	382657176	763	582169	444194947
727	528529	384240583	764	583696	445943774
728	529984	385828352	765	585225	447697125
729	531441	387420489	766	586756	449455096
			767	588289	451217663
730	532900	389017000	768	589824	452914832
731	534361	390617891	769	591361	454756609
732	535824	392223168			
733	537289	393832837	770	592900	456533000
734	538756	395446904	771	594441	458314011
735	540225	397065375	772	595984	460099648
736	541696	398688256	773	597529	461889917
737	543169	400315553	774	599076	463684824
738	544644	401947272	775	600625	465484375
739	546121	403583419	776	602176	467288576
			777	603729	469097433
740	547600	405224000	778	605284	470910952
741	549081	406869021	779	606841	472729139
742	550564	408518488			
743	552049	410172407	780	608400	474552000
744	553536	411830784	781	609961	476379541
745	555025	413493625	782	611524	478211768
746	556516	415160936	783	613089	480048687
747	558009	416832723	784	614656	481890304
748	559504	418508992	785	616225	483736025
749	561001	420189749	786	617796	485587656
			787	619369	487443403
750	562500	421875000	788	620944	489303872
751	564001	423564751	789	622521	491169069
752	565504	424525900			
753	567009	426957777	790	624100	493039000
754	568516	428661064	791	625681	494913671
755	570025	430368875	792	627264	496793088
756	571536	432081216	793	628849	498677257
757	573049	433798093	794	630436	500566184
758	574564	435519512	795	632025	502459875
759	576081	437245479	796	633616	504358336

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
797	635209	506261573	833	693889	578009537
798	636804	508169592	834	695556	580093704
799	638401	510082399	835	697225	582182875
			836	698896	584277056
800	640000	512000000	837	700569	586376253
801	641601	513922401	838	702244	588480472
802	643204	515849608	839	703921	590589719
803	644809	517781627			
804	646416	519718464	840	705600	592704000
805	648025	521660125	841	707281	594823321
806	649636	523606616	842	708964	596947688
807	651249	525557943	843	710649	599077107
808	652864	527514112	844	712336	601211584
809	654481	529475129	845	714025	603351125
			846	715716	605495736
810	656100	531441000	847	717409	607645423
811	657721	533411731	848	719104	609800192
812	659344	535387328	849	720801	611960049
813	660969	537366797			
814	662596	539353144	850	722500	614125000
815	664225	541343375	851	724201	616295051
816	665856	543338496	852	725904	618470208
817	667489	545338513	853	727609	620650477
818	669124	547343432	854	729316	622835864
819	670761	549353259	855	731025	625026375
			856	732736	627222016
820	672400	551368000	857	734449	629422793
821	674041	553387661	858	736164	631628712
822	675684	555412248	859	737881	633839779
823	677329	557441767			
824	678976	559476224	860	739600	636056000
825	680625	561515625	861	741321	638277381
826	682276	563559976	862	743044	640503928
827	683929	565609283	863	744769	642735647
828	685584	567663552	864	746496	644972544
829	687241	569722789	865	748225	647214625
			866	749956	649461896
830	688900	571787000	867	751689	651714363
831	690561	573856191	868	753424	653972032
832	692224	575930368	869	755161	656234909

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
870	756900	658503000	907	822649	746142643
871	758641	660776311	908	824464	748613312
872	760384	663054848	909	826281	751089429
873	762129	665338617			
874	763876	667627624	910	828100	753571000
875	765625	669921875	911	829921	756058031
876	767376	672221376	912	831744	758550528
877	769129	674526133	913	833569	761048497
878	770884	676836152	914	835396	763551944
879	772641	679151439	915	837225	766060875
			916	839056	768575296
880	774400	681472000	917	840889	771095213
881	776161	683797841	918	842724	773620632
882	777924	686128968	919	844561	776151559
883	779689	688465387			
884	781456	690807104	920	846400	778688000
885	783225	693154125	921	848241	781229961
886	784996	695506456	922	850084	783777448
887	786769	697864103	923	851929	786330467
888	788544	700227072	924	853776	788889024
889	790321	702595369	925	855625	791453125
			926	857476	794022776
890	792100	704969000	927	859329	796597983
891	793881	707347971	928	861184	799178752
892	795664	709732288	929	863041	801765889
893	797449	712121957			
894	799236	714516984	930	864900	804375000
895	801025	716917375	931	866761	806954491
896	802816	719323136	932	868624	809557568
897	804609	721734273	933	870489	812166237
898	806404	724150792	934	872356	814780504
899	808201	726572699	935	874225	817400375
			936	876096	820025856
900	810000	729000000	937	877969	822656953
901	811804	731432701	938	879844	825293672
902	813604	733870808	939	881721	827936019
903	815409	736314327			
904	817216	738763264	940	883600	830584000
905	819025	741217625	941	885481	833237621
906	820836	743677416	942	887364	835896888

TABLE I (*Continued*).

Number. N	Square. N^2	Cube. N^3	Number. N	Square. N^2	Cube. N^3
943	889249	838561807	972	944784	918330048
944	891136	841232384	973	946729	921167317
945	893025	843908625	974	948676	924010424
946	894916	846590536	975	950625	926859375
947	896809	849278123	976	952576	929714176
948	898704	851971392	977	954529	932574833
949	900601	854670349	978	956484	935441352
			979	958441	938313739
950	902500	857375000			
951	904401	860085351	980	960400	941192000
952	906304	862801408	981	962361	944076141
953	908209	865523177	982	964324	946966168
954	910116	868250664	983	966289	949862087
955	912025	870983875	984	968256	952763904
956	913936	873722816	985	970225	955671625
957	915849	876467493	986	972196	958585256
958	917764	879217912	987	974169	961504803
959	919681	881974079	988	976144	964430272
			989	978121	967361669
960	921600	884736000			
961	923521	887503681	990	980100	970299000
962	925444	890277128	991	982081	973242271
963	927369	893056347	992	984064	976191488
964	929296	895841344	993	986049	979146657
965	931225	898632125	994	988036	982107784
966	933156	901428696	995	990025	985074875
967	935089	904231063	996	992016	988047936
968	937024	907039232	997	994009	991026973
969	938961	909853209	998	996004	994011992
			999	998001	997002999
970	940900	912673000			
971	942841	915498611	1000	1000000	1000000000

Table 2.

Fourth and Fifth Powers of Numbers.

No.	N^4	N^5	No.	N^4	N^5
⅛	·00024	·000031	7¾	3607·504	27958·155
¼	·00391	·000977	8	4096·000	32768·000
⅜	·01978	·007416	8¼	4632·504	38218·157
½	·06250	·031250	8½	5220·062	44370·531
⅝	·15259	·095367	8¾	5861·816	51290·894
¾	·31641	·237305	9	6561·000	59049·000
⅞	·58619	·512909	9¼	7320·941	67718·708
1	1·00000	1·000000	9½	8145·062	77378·094
1⅛	1·60181	1·802032	9¾	9036·879	88109·669
1¼	2·44141	3·051758	10	10000·000	100000·
1⅜	3·57446	4·914886	10¼	11038·129	113140·821
1½	5·06250	7·593750	10½	12155·062	127628·156
1⅝	6·97290	11·33096	10¾	13354·691	143562·933
1¾	9·37891	16·41308	11	14641·000	161051·000
1⅞	12·35962	23·17428	11¼	16018·066	180203·247
2	16·00000	32·0000	11½	17490·062	201135·719
2⅛	20·39087	43·33059	11¾	19061·253	223969·733
2¼	25·62891	57·66504	12	20736·000	248832·000
2⅜	31·81665	75·56454	12¼	22518·753	275854·735
2½	39·06250	97·65625	12½	24414·062	305175·781
2⅝	47·48071	124·6369	12¾	26426·566	336938·722
2¾	57·19141	154·2764	13	28561·000	371293·000
2⅞	68·32056	194·4216	13¼	30822·191	408374·036
3	81·00000	243·0000	13½	33215·062	448403·344
3¼	111·5664	362·5908	13¾	35744·629	491488·647
3½	150·0625	525·2187	14	38416·000	537824·000
3¾	197·7539	741·5771	14¼	41234·378	587589·899
4	256·0000	1024·000	14½	44205·062	640973·406
4¼	326·2539	1386·579	14¾	47333·441	698168·261
4½	410·0625	1845·281	15	50625·000	759375·000
4¾	509·0664	2418·065	15¼	54085·316	824801·075
5	625·0000	3125·000	15½	57720·062	894660·969
5¼	759·6914	3988·379	15¾	61535·004	969176·312
5½	915·0625	5032·843	16	65536·000	1048576·000
5¾	1093·129	6285·491	16¼	69729·004	1133096·313
6	1296·000	7776·000	16½	74120·062	1222981·031
6¼	1525·879	9536·743	16¾	78715·316	1318481·549
6½	1785·062	11602·906	17	83521·000	1419857·000
6¾	2075·941	14012·604	17¼	88543·441	1527374·364
7	2401·000	16807·000	17½	93789·062	1641308·593
7¼	2762·816	20030·418	17¾	99264·379	1761942·726
7½	3164·062	23750·468	18	104976·000	1889568·000

TABLE 2 (*Continued*).

No.	N^4	N^5	No.	N^4	N^5
18¼	110930·64	2024284·325	37½	1977539·06	74157714·84
18½	117727·062	2177950·656	38	2085136·00	79235168·00
18¾	123596·191	2317428·589	38½	2197065·06	84587004·91
19	130321·000	2476099·000	39	2313441·00	90224199·00
19¼	137316·566	2643343·903	39½	2434380·06	96158012·47
19½	144590·062	2819506·219	40	2560000·00	102400000·0
19¾	152148·754	3004937·889	40½	2690420·06	108962012·5
20	160000·000	3200000·000	41	2825761·00	115856201·0
20½	176610·062	3720506·281	41½	2966145·06	123095020·1
21	194481·000	4084101·000	42	3111696·00	130691232·0
21½	213675·062	4594013·844	42½	3262539·06	138657910·2
22	234256·000	5153632·000	43	3418801·00	147008443·0
22½	256289·062	5766503·906	43½	3580610·06	155756537·7
23	279841·000	6436343·000	44	3748096·00	164916224·0
23½	304980·062	7167031·469	44½	3921390·06	174501857·8
24	331776·000	7962624·000	45	4100625·00	184528125·0
24½	360300·062	8827351·531	45½	4285935·06	195010045·3
25	390625·000	9765625·00	46	4477456·00	205962976·0
25½	422825·062	10782039·09	46½	4675325·06	217402615·4
26	456976·000	11881376·00	47	4879681·00	229345007·0
26½	493155·062	13068609·16	47½	5090664·06	241806542·9
27	531441·000	14348907·00	48	5308416·00	254803968·0
27½	571914·062	15727636·72	48½	5533080·06	268354383·0
28	614656·000	17210368·00	49	5764801·00	282475249·0
28½	659750·062	18802876·78	49½	6003725·06	297184390·6
29	707281·000	20511149·00	50	6250000·00	312500000·0
29½	757335·062	22341384·34	50½	6503775·06	328440640·7
30	810000·000	24300000·00	51	6765201·00	345025251·0
30½	865365·062	26393637·41	51½	7034430·06	362273148·2
31	923521·000	28629151·00	52	7311616·00	380204032·0
31½	984560·062	31013641·97	52½	7596914·06	398837988·3
32	1048576·00	33554432·00	53	7890481·00	418195493·0
32½	1115664·06	36259082·03	53½	8192475·06	438297415·8
33	1185921·00	39135393·00	54	8503056·00	459165024·0
33½	1259445·06	42191409·59	54½	8822385·06	480819985·9
34	1336336·00	45435424·00	55	9150625·00	503284375·0
34½	1416695·06	48875979·66	55½	9487940·06	526580673·5
35	1500625·00	52521875·00	56	9834496·00	550731776·0
35½	1589230·06	56417677·22	56½	10190460·1	575760993·5
36	1679616·00	60466176·00	57	10556001·0	601692057·0
36½	1774890·06	64783487·28	57½	10931289·1	628549121·1
37	1874161·00	69343957·00	58	11316496·0	656356768·0

TABLE 2 (*Continued*).

No.	N^4	N^5	No.	N^4	N^5
58½	11711795·1	685140011·2	79½	40008762	3180696623
59	12117361·0	714924299·0	80	40960000	3276800000
59½	12533370·0	745735518·7	80½	41993640	3380488025
60	12960000	777600000·0	81	43046721	3486784401
60½	13397430	810544518·8	81½	44119485	3595738032
61	13845841	844596301·0	82	45212176	3707398432
61½	14305415	879783026·3	82½	46325039	3821815722
62	14776336	916132832·0	83	47458321	3939040643
62½	15258789	953674316·4	83½	48612270	4059124550
63	15752961	992436543·0	84	49787136	4182119424
63½	16259040	1032449043	84½	50983170	4308077870
64	16777216	1073741824	85	52055125	4424685625
64½	17307680	1116345364	85½	53439755	4569099057
65	17850625	1160290625	86	54700816	4704270176
65½	18406245	1205609051	86½	55984065	4842621627
66	18974736	1252332576	87	57289761	4984209207
66½	19556295	1300493621	87½	58618164	5129089355
67	20151121	1350125107	88	59969536	5277319168
67½	20759414	1401260449	88½	61344140	5428956395
68	21381376	1453933568	89	62742241	5584059449
68½	22017210	1508178889	89½	64164105	5742687403
69	22667121	1564031349	90	65610000	5904900000
69½	23331315	1621526396	90½	67080195	6070757653
70	24010000	1680700000	91	68574961	6240321451
70½	24703385	1741588646	91½	70094570	6413653160
71	25411681	1804229351	92	71639296	6590815232
71½	26135100	1868659654	92½	73209414	6771870800
72	26873856	1934917632	93	74805201	6956883693
72½	27628164	2003041894	93½	76426935	7145918428
73	28398241	2073071593	94	78074896	7339040224
73½	29184305	2145046422	94½	79749365	7536314998
74	29986576	2219006624	95	81450625	7737809375
74½	30805275	2294992992	95½	83178950	7943589685
75	31640625	2373046875	96	84934656	8153726976
75½	32492850	2453210179	96½	86811122	8377273327
76	33362176	2535525376	97	88529281	8587340257
76½	34248830	2620035499	97½	90368789	8810956933
77	35153041	2706784157	98	92236816	9039207968
77½	36075039	2795815527	98½	94133655	9272155023
78	37015056	2887174368	99	96059601	9509900499
78½	37973325	2980906017	99½	98014950	9752487531
79	38950081	3077056399	100	100000000	10000000000

TABLE 2 (*Continued*).

No.	N^4	N^5	No.	N^4	N^5
101	104060401	10510100501	126	252047376	31757969376
102	108243216	11040808032	127	260144641	33038369407
103	112550881	11592740743	128	268435456	34359738368
104	116985856	12166529024	129	276922881	35723051649
105	121550625	12762815625	130	285610000	37129300000
106	126247696	13382255776	131	294499921	38579489651
107	131079601	14025517307	132	303595776	40074642432
108	136048896	14693280768	133	312900721	41615795893
109	141158161	15386239549	134	322417936	43204003424
110	146410000	16105100000	135	332150625	44840334375
111	151807041	16850581551	136	342102016	46525874176
112	157351936	17623416832	137	352275361	48261724457
113	163047361	18424351793	138	362673936	50049003168
114	168896016	19254145824	139	373301041	51888844699
115	174900625	20113571875	140	384160000	53782400000
116	181063936	21003416576	141	395254161	55730836701
117	187388721	21924480357	142	406586896	57735339232
118	193877776	22877577568	143	418161601	59797108943
119	200533921	23863536599	144	429981696	61917364224
120	207360000	24883200000	145	442050625	64097340625
121	214358881	25937424601	146	454371856	66338290976
122	221533456	27027081632	147	466948881	68641485507
123	228886641	28153056843	148	479785216	71008211968
124	236421376	29316250624	149	492884401	73439775749
125	244140625	30517578125	150	506250000	75937500000

TABLE 3.

SQUARE AND CUBE ROOTS.

No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$
1	1·0000	1·0000	22	4·6904	2·8020	43	6·5574	3·5034
1½	1·2071	1·1299	22½	4·7431	2·8230	43½	6·5953	3·5169
2	1·4142	1·2599	23	4·7958	2·8439	44	6·6332	3·5303
2½	1·5731	1·3511	23½	4·8474	2·8642	44½	6·6707	3·5436
3	1·7321	1·4422	24	4·8990	2·8845	45	6·7082	3·5569
3½	1·8660	1·5149	24½	4·9495	2·9043	45½	6·7453	3·5700
4	2·0000	1·5874	25	5·0000	2·9240	46	6·7823	3·5830
4½	2·1180	1·6487	25½	5·0495	2·9433	46½	6·8189	3·5959
5	2·2361	1·7100	26	5·0990	2·9625	47	6·8557	3·6088
5½	2·3428	1·7635	26½	5·1476	2·9813	47½	6·8919	3·6215
6	2·4495	1·8171	27	5·1962	3·0000	48	6·9282	3·6342
6½	2·5476	1·8650	27½	5·2438	3·0183	48½	6·9641	3·6468
7	2·6458	1·9129	28	5·2915	3·0366	49	7·0000	3·6593
7½	2·7371	1·9565	28½	5·3383	3·0545	49½	7·0355	3·6717
8	2·8284	2·0000	29	5·3852	3·0723	50	7·0711	3·6840
8½	2·9142	2·0401	29½	5·4312	3·0898	50½	7·1063	3·6962
9	3·0000	2·0801	30	5·4772	3·1072	51	7·1414	3·7084
9½	3·0811	2·1173	30½	5·5224	3·1243	51½	7·1763	3·7205
10	3·1632	2·1544	31	5·5678	3·1414	52	7·2111	3·7325
10½	3·2395	2·1892	31½	5·6123	3·1581	52½	7·2456	3·7444
11	3·3166	2·2240	32	5·6569	3·1748	53	7·2801	3·7563
11½	3·3904	2·2567	32½	5·7007	3·1912	53½	7·3143	3·7681
12	3·4641	2·2894	33	5·7446	3·2075	54	7·3485	3·7798
12½	3·5348	2·3204	33½	5·7878	3·2236	54½	7·3823	3·7914
13	3·6056	2·3513	34	5·8310	3·2396	55	7·4162	3·8030
13½	3·6736	2·3807	34½	5·8735	3·2554	55½	7·4498	3·8145
14	3·7414	2·4101	35	5·9161	3·2711	56	7·4833	3·8259
14½	3·8073	2·4382	35½	5·9580	3·2865	56½	7·5166	3·8372
15	3·8730	2·4662	36	6·0000	3·3019	57	7·5498	3·8485
15½	3·9365	2·4930	36½	6·0414	3·3171	57½	7·5828	3·8597
16	4·0000	2·5198	37	6·0828	3·3322	58	7·6158	3·8709
16½	4·0616	2·5456	37½	6·1236	3·3471	58½	7·6485	3·8820
17	4·1231	2·5713	38	6·1644	3·3620	59	7·6811	3·8930
17½	4·1829	2·5960	38½	6·2047	3·3766	59½	7·7136	3·9040
18	4·2426	2·6207	39	6·2450	3·3912	60	7·7460	3·9149
18½	4·3008	2·6446	39½	6·2848	3·4056	60½	7·7781	3·9257
19	4·3589	2·6684	40	6·3246	3·4200	61	7·8102	3·9365
19½	4·4155	2·6914	40½	6·3638	3·4341	61½	7·8421	3·9472
20	4·4721	2·7144	41	6·4031	3·4482	62	7·8740	3·9579
20½	4·5274	2·7367	41½	6·4419	3·4621	62½	7·9056	3·9685
21	4·5826	2·7589	42	6·4807	3·4760	63	7·9373	3·9791
21½	4·6365	2·7805	42½	6·5191	3·4897	63½	7·9686	3·9896

TABLE 3 (*Continued*).

$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$
8·0000	4·0000	85	9·2195	4·3968	106	10·295	4·7326
8·0311	4·0104	85½	9·2466	4·4054	106½	10·319	4·7401
8·0623	4·0207	86	9·2736	4·4140	107	10·344	4·7475
8·0932	4·0310	86½	9·3005	4·4225	107½	10·368	4·7549
8·1240	4·0412	87	9·3274	4·4310	108	10·392	4·7622
8·1547	4·0514	87½	9·3541	4·4395	108½	10·416	4·7695
8·1854	4·0615	88	9·3808	4·4480	109	10·440	4·7769
8·2158	4·0716	88½	9·4074	4·4563	109½	10·464	4·7842
8·2462	4·0817	89	9·4340	4·4647	110	10·488	4·7914
8·2764	4·0917	89½	9·4604	4·4731	110½	10·511	4·7987
8·3066	4·1016	90	9·4868	4·4814	111	10·538	4·8059
8·3366	4·1115	90½	9·5131	4·4897	111½	10·559	4·8131
8·3666	4·1213	91	9·5394	4·4979	112	10·583	4·8203
8·3964	4·1311	91½	9·5655	4·5062	112½	10·606	4·8275
8·4261	4·1408	92	9·5917	4·5144	113	10·630	4·8346
8·4557	4·1505	92½	9·6177	4·5226	113½	10·653	4·8417
8·4853	4·1602	93	9·6437	4·5307	114	10·677	4·8488
8·5146	4·1698	93½	9·6695	4·5388	114½	10·700	4·8559
8·5440	4·1793	94	9·6954	4·5468	115	10·723	4·8629
8·5732	4·1888	94½	9·7211	4·5549	115½	10·747	4·8699
8·6023	4·1983	95	9·7468	4·5629	116	10·770	4·8770
8·6313	4·2078	95½	9·7724	4·5709	116½	10·793	4·8840
8·6603	4·2172	96	9·7980	4·5789	117	10·816	4·8910
8·6890	4·2265	96½	9·8234	4·5868	117½	10·839	4·8979
8·7178	4·2358	97	9·8489	4·5947	118	10·862	4·9049
8·7464	4·2451	97½	9·8742	4·6026	118½	10·885	4·9118
8·7750	4·2543	98	9·8995	4·6104	119	10·908	4·9187
8·8034	4·2635	98½	9·9247	4·6183	119½	10·931	4·9256
8·8318	4·2727	99	9·9499	4·6261	120	10·954	4·9324
8·8600	4·2818	99½	9·9749	4·6339	120½	10·977	4·9393
8·8882	4·2908	100	10·000	4·6416	121	11·000	4·9461
8·9162	4·2999	100½	10·024	4·6493	121½	11·022	4·9529
8·9443	4·3089	101	10·049	4·6570	122	11·045	4·9597
8·9722	4·3178	101½	10·074	4·6647	122½	11·068	4·9665
9·0000	4·3267	102	10·099	4·6723	123	11·090	4·9732
9·0279	4·3356	102½	10·124	4·6799	123½	11·113	4·9799
9·0554	4·3445	103	10·148	4·6875	124	11·135	4·9866
9·0829	4·3533	103½	10·173	4·6951	124½	11·158	4·9933
9·1104	4·3621	104	10·198	4·7027	125	11·180	5·0000
9·1378	4·3708	104½	10·222	4·7102	125½	11·202	5·0067
9·1652	4·3795	105	10·247	4·7177	126	11·225	5·0133
9·1924	4·3382	105½	10·271	4·7252	126½	11·247	5·0199

TABLE 3 (*Continued*).

No.	√N	∛N	No.	√N	∛N	No.	√N	∛N
127	11·269	5·0265	148	12·165	5·2896	169	13·000	5·5288
127½	11·291	5·0331	148½	12·186	5·2956	169½	13·019	5·5343
128	11·313	5·0397	149	12·206	5·3015	170	13·038	5·5397
128½	11·335	5·0462	149½	12·227	5·3074	170½	13·057	5·5451
129	11·357	5·0528	150	12·247	5·3133	171	13·076	5·5505
129½	11·379	5·0559	150½	12·267	5·3192	171½	13·095	5·5559
130	11·401	5·0658	151	12·288	5·3251	172	13·114	5·5613
130½	11·423	5·0720	151½	12·308	5·3310	172½	13·133	5·5667
131	11·445	5·0788	152	12·328	5·3368	173	13·152	5·5721
131½	11·467	5·0852	152½	12·349	5·3427	173½	13·171	5·5775
132	11·489	5·0916	153	12·369	5·3485	174	13·190	5·5828
132½	11·510	5·0981	153½	12·389	5·3540	174½	13·209	5·5883
133	11·532	5·1045	154	12·409	5·3601	175	13·228	5·5938
133½	11·554	5·1109	154½	12·429	5·3659	175½	13·247	5·5990
134	11·575	5·1172	155	12·449	5·3717	176	13·266	5·6041
134½	11·597	5·1236	155½	12·470	5·3773	176½	13·285	5·6094
135	11·619	5·1299	156	12·490	5·3832	177	13·304	5·6147
135½	11·640	5·1363	156½	12·510	5·3890	177½	13·322	5·6200
136	11·661	5·1426	157	12·530	5·3947	178	13·341	5·6252
136½	11·683	5·1489	157½	12·550	5·4004	178½	13·360	5·6305
137	11·704	5·1551	158	12·569	5·4061	179	13·379	5·6357
137½	11·726	5·1614	158½	12·589	5·4118	179½	13·397	5·6410
138	11·747	5·1676	159	12·609	5·4175	180	13·416	5·6462
138½	11·768	5·1739	159½	12·629	5·4232	180½	13·435	5·6515
139	11·789	5·1801	160	12·649	5·4288	181	13·453	5·6567
139½	11·811	5·1863	160½	12·668	5·4345	181½	13·472	5·6619
140	11·832	5·1925	161	12·688	5·4401	182	13·490	5·6671
140½	11·853	5·1987	161½	12·708	5·4458	182½	13·509	5·6723
141	11·874	5·2048	162	12·727	5·4514	183	13·527	5·6774
141½	11·895	5·2110	162½	12·747	5·4570	183½	13·546	5·6826
142	11·916	5·2171	163	12·767	5·4626	184	13·564	5·6877
142½	11·937	5·2232	163½	12·786	5·4682	184½	13·583	5·6929
143	11·958	5·2293	164	12·806	5·4737	185	13·601	5·6980
143½	11·979	5·2354	164½	12·825	5·4793	185½	13·619	5·7032
144	12·000	5·2415	165	12·845	5·4848	186	13·638	5·7083
144½	12·020	5·2476	165½	12·864	5·4904	186½	13·656	5·7134
145	12·041	5·2536	166	12·884	5·4959	187	13·674	5·7185
145½	12·062	5·2596	166½	12·903	5·5014	187½	13·693	5·7236
146	12·083	5·2656	167	12·922	5·5069	188	13·711	5·7287
146½	12·103	5·2716	167½	12·942	5·5124	188½	13·729	5·7338
147	12·124	5·2776	168	12·961	5·5178	189	13·747	5·7388
147½	12·145	5·2836	168½	12·980	5·5233	189½	13·765	5·7439

TABLE 3 (*Continued*).

No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$
190	13·784	5·7489	222	14·899	6·0550	264	16·248	6·415
190½	13·802	5·7540	223	14·933	6·0641	265	16·278	6·423
191	13·820	5·7590	224	14·966	6·0732	266	16·309	6·431
191½	13·838	5·7640	225	15·000	6·0822	267	16·340	6·439.
192	13·856	5·7690	226	15·033	6·0912	268	16·370	6·447
192½	13·874	5·7740	227	15·066	6·1002	269	16·401	6·455
193	13·892	5·7790	228	15·099	6·1091	270	16·431	6·463
193½	13·910	5·7840	229	15·132	6·1180	271	16·462	6·471
194	13·928	5·7890	230	15·165	6·1269	272	16·492	6·479
194½	13·946	5·7940	231	15·198	6·1358	273	16·522	6·487
195	13·964	5·7989	232	15·231	6·1446	274	16·552	6·495
195½	13·982	5·8039	233	15·264	6·1534	275	16·583	6·502
196	14·000	5·8088	234	15·297	6·1622	276	16·613	6·510
196½	14·017	5·8137	235	15·329	6·1710	277	16·643	6·518
197	14·035	5·8186	236	15·362	6·1797	278	16·673	6·526
197½	14·053	5·8236	237	15·394	6·1885	279	16·703	6·534
198	14·071	5·8285	238	15·427	6·1972	280	16·733	6·542
198½	14·089	5·8334	239	15·459	6·2058	281	16·763	6·549
199	14·106	5·8383	240	15·491	6·2145	282	16·792	6·557
199½	14·124	5·8432	241	15·524	6·2231	283	16·822	6·565
200	14·142	5·8480	242	15·556	6·2317	284	16·852	6·573
201	14·177	5·8578	243	15·588	6·2403	285	16·881	6·580
202	14·212	5·8675	244	15·620	6·2488	286	16·911	6·588
203	14·247	5·8771	245	15·652	6·2573	287	16·941	6·596
204	14·282	5·8868	246	15·684	6·2658	288	16·970	6·603
205	14·317	5·8964	247	15·716	6·2743	289	17·000	6·611
206	14·352	5·9059	248	15·748	6·2828	290	17·029	6·619
207	14·387	5·9155	249	15·779	6·2912	291	17·059	6·627
208	14·422	5·9250	250	15·811	6·2996	292	17·088	6·634
209	14·456	5·9845	251	15·843	6·3080	293	17·117	6·642
210	14·491	5·9439	252	15·874	6·3164	294	17·146	6·649
211	14·525	5·9533	253	15·906	6·3247	295	17·176	6·657
212	14·560	5·9627	254	15·937	6·3330	296	17·205	6·664
213	14·594	5·9721	255	15·968	6·3413	297	17·234	6·672
214	14·628	5·9814	256	16·000	6·3496	298	17·263	6·679
215	14·662	5·9907	257	16·031	6·3579	299	17·292	6·687
216	14·696	6·0000	258	16·062	6·3661	300	17·320	6·694
217	14·730	6·0092	259	16·093	6·3743	301	17·349	6·702
218	14·764	6·0185	260	16·124	6·3825	302	17·378	6·709
219	14·798	6·0277	261	16·155	6·3907	303	17·407	6·717
220	14·832	6·0368	262	16·186	6·3988	304	17·436	6·724
221	14·866	6·0459	263	16·217	6·4070	305	17·464	6·731

TABLE 3 (*Continued*).

No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$
306	17·493	6·739	348	18·655	7·034	390	19·748	7·306
307	17·521	6·746	349	18·681	7·040	391	19·774	7·312
308	17·549	6·753	350	18·708	7·047	392	19·799	7·319
309	17·578	6·761	351	18·735	7·054	393	19·824	7·325
310	17·607	6·768	352	18·762	7·061	394	19·849	7·331
311	17·635	6·775	353	18·788	7·067	395	19·875	7·337
312	17·663	6·782	354	18·815	7·074	396	19·899	7·343
313	17·692	6·789	355	18·842	7·081	397	19·925	7·349
314	17·720	6·797	356	18·868	7·087	398	19·949	7·356
315	17·748	6·804	357	18·894	7·094	399	19·975	7·362
316	17·776	6·811	358	18·921	7·101	400	20·000	7·368
317	17·804	6·818	359	18·947	7·107	401	20·025	7·374
318	17·832	6·826	360	18·974	7·114	402	20·049	7·380
319	17·860	6·833	361	19·000	7·120	403	20·075	7·386
320	17·888	6·839	362	19·026	7·127	404	20·099	7·392
321	17·916	6·847	363	19·052	7·133	405	20·125	7·399
322	17·944	6·854	364	19·079	7·140	406	20·149	7·405
323	17·972	6·861	365	19·105	7·146	407	20·174	7·411
324	18·000	6·868	366	19·131	7·153	408	20·199	7·417
325	18·028	6·875	367	19·157	7·159	409	20·224	7·422
326	18·055	6·882	368	19·183	7·166	410	20·248	7·429
327	18·083	6·889	369	19·209	7·172	411	20·273	7·434
328	18·111	6·896	370	19·235	7·179	412	20·298	7·441
329	18·138	6·903	371	19·261	7·185	413	20·322	7·447
330	18·166	6·910	372	19·287	7·192	414	20·347	7·453
331	18·193	6·917	373	19·313	7·198	415	20·371	7·459
332	18·221	6·924	374	19·339	7·205	416	20·396	7·465
333	18·248	6·931	375	19·365	7·211	417	20·421	7·471
334	18·276	6·938	376	19·391	7·218	418	20·445	7·477
335	18·303	6·945	377	19·416	7·224	419	20·469	7·483
336	18·330	6·952	378	19·442	7·230	420	20·494	7·489
337	18·357	6·959	379	19·468	7·237	421	20·518	7·495
338	18·385	6·966	380	19·493	7·243	422	20·543	7·501
339	18·412	6·973	381	19·519	7·249	423	20·567	7·507
340	18·439	6·979	382	19·545	7·256	424	20·591	7·513
341	18·466	6·986	383	19·570	7·262	425	20·615	7·518
342	18·493	6·993	384	19·596	7·268	426	20·639	7·524
343	18·520	7·000	385	19·621	7·275	427	20·664	7·530
344	18·547	7·007	386	19·647	7·281	428	20·688	7·536
345	18·574	7·014	387	19·672	7·287	429	20·712	7·542
346	18·601	7·020	388	19·698	7·294	430	20·736	7·548
347	18·628	7·027	389	19·723	7·299	431	20·760	7·554

TABLE 3 (*Continued*).

No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$
432	20·785	7·559	474	21·771	7·797	516	22·716	8·021
433	20·809	7·565	475	21·794	7·802	517	22·738	8·026
434	20·833	7·571	476	21·817	7·808	518	22·759	8·031
435	20·857	7·577	477	21·840	7·813	519	22·782	8·036
436	20·881	7·583	478	21·863	7·819	520	22·803	8·041
437	20·904	7·588	479	21·886	7·824	521	22·825	8·047
438	20·928	7·594	480	21·909	7·830	522	22·847	8·052
439	20·952	7·600	481	21·932	7·835	523	22·869	8·057
440	20·976	7·606	482	21·954	7·840	524	22·891	8·062
441	21·000	7·612	483	21·977	7·846	525	22·913	8·067
442	21·024	7·617	484	22·000	7·851	526	22·935	8·072
443	21·047	7·623	485	22·023	7·857	527	22·956	8·077
444	21·071	7·629	486	22·045	7·862	528	22·978	8·082
445	21·095	7·635	487	22·069	7·868	529	23·000	8·087
446	21·119	7·640	488	22·091	7·873	530	23·022	8·093
447	21·142	7·646	489	22·113	7·878	531	23·043	8·098
448	21·166	7·652	490	22·136	7·884	532	23·065	8·103
449	21·189	7·657	491	22·158	7·889	533	23·087	8·108
450	21·213	7·663	492	22·181	7·894	534	23·108	8·113
451	21·237	7·669	493	22·204	7·899	535	23·130	8·118
452	21·260	7·674	494	22·226	7·905	536	23·152	8·123
453	21·284	7·680	495	22·248	7·910	537	23·173	8·128
454	21·307	7·686	496	22·271	7·915	538	23·195	8·133
455	21·331	7·691	497	22·293	7·921	539	23·216	8·138
456	21·354	7·697	498	22·316	7·926	540	23·238	8·143
457	21·377	7·703	499	22·338	7·932	541	23·259	8·148
458	21·401	7·708	500	22·361	7·937	542	23·281	8·153
459	21·424	7·714	501	22·383	7·942	543	23·302	8·158
460	21·447	7·719	502	22·405	7·947	544	23·324	8·163
461	21·471	7·725	503	22·428	7·953	545	23·345	8·168
462	21·494	7·731	504	22·449	7·958	546	23·367	8·173
463	21·517	7·736	505	22·472	7·963	547	23·388	8·178
464	21·541	7·742	506	22·494	7·969	548	23·409	8·183
465	21·564	7·747	507	22·517	7·974	549	23·431	8·188
466	21·587	7·753	508	22·539	7·979	550	23·452	8·193
467	21·610	7·758	509	22·561	7·984	551	23·473	8·198
468	21·633	7·764	510	22·583	7·989	552	23·495	8·203
469	21·656	7·769	511	22·605	7·995	553	23·516	8·208
470	21·679	7·775	512	22·627	8·000	554	23·537	8·213
471	21·702	7·780	513	22·649	8·005	555	23·558	8·218
472	21·725	7·786	514	22·671	8·010	556	23·579	8·223
473	21·749	7·791	515	22·694	8·016	557	23·601	8·228

TABLE 3 (*Continued*).

No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$
558	23·622	8·233	600	24·495	8·434	642	25·338	8·627
559	23·643	8·238	601	24·515	8·439	643	25·357	8·631
560	23·664	8·242	602	24·536	8·444	644	25·377	8·636
561	23·685	8·247	603	24·556	8·448	645	25·397	8·640
562	23·706	8·252	604	24·576	8·453	646	25·416	8·644
563	23·728	8·257	605	24·597	8·458	647	25·436	8·649
564	23·749	8·262	606	24·617	8·462	648	25·456	8·653
565	23·769	8·267	607	24·637	8·467	649	25·475	8·658
566	23·791	8·272	608	24·658	8·472	650	25·495	8·662
567	23·812	8·277	609	24·678	8·476	651	25·515	8·667
568	23·833	8·282	610	24·698	8·481	652	25·534	8·671
569	23·854	8·286	611	24·718	8·485	653	25·554	8·676
570	23·875	8·291	612	24·739	8·490	654	25·573	8·680
571	23·896	8·296	613	24·758	8·495	655	25·593	8·684
572	23·916	8·301	614	24·779	8·499	656	25·612	8·689
573	23·937	8·306	615	24·799	8·504	657	25·632	8·693
574	23·958	8·311	616	24·819	8·509	658	25·651	8·698
575	23·979	8·315	617	24·839	8·513	659	25·671	8·702
576	24·000	8·320	618	24·859	8·518	660	25·690	8·706
577	24·021	8·325	619	24·879	8·522	661	25·710	8·711
578	24·042	8·330	620	24·899	8·527	662	25·730	8·715
579	24·062	8·335	621	24·919	8·532	663	25·749	8·719
580	24·083	8·339	622	24·939	8·536	664	25·768	8·724
581	24·104	8·344	623	24·959	8·541	665	25·787	8·728
582	24·125	8·349	624	24·980	8·545	666	25·807	8·733
583	24·145	8·354	625	25·000	8·549	667	25·826	8·737
584	24·166	8·359	626	25·019	8·554	668	25·846	8·742
585	24·187	8·363	627	25·040	8·559	669	25·865	8·746
586	24·207	8·368	628	25·059	8·563	670	25·884	8·750
587	24·228	8·373	629	25·079	8·568	671	25·904	8·755
588	24·249	8·378	630	25·099	8·573	672	25·923	8·759
589	24·269	8·382	631	25·119	8·577	673	25·942	8·763
590	24·289	8·387	632	25·139	8·582	674	25·961	8·768
591	24·310	8·392	633	25·159	8·586	675	25·981	8·772
592	24·331	8·397	634	25·179	8·591	676	26·000	8·776
593	24·351	8·401	635	25·199	8·595	677	26·019	8·781
594	24·372	8·406	636	25·219	8·599	678	26·038	8·785
595	24·393	8·411	637	25·239	8·604	679	26·058	8·789
596	24·413	8·415	638	25·259	8·609	680	26·077	8·794
597	24·433	8·420	639	25·278	8·613	681	26·096	8·798
598	24·454	8·425	640	25·298	8·618	682	26·115	8·802
599	24·474	8·429	641	25·318	8·622	683	26·134	8·807

TABLE 3 (*Continued*).

No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$
684	26·153	8·811	726	26·944	8·988	768	27·713	9·158
685	26·172	8·815	727	26·963	8·992	769	27·731	9·162
686	26·192	8·819	728	26·981	8·996	770	27·749	9·166
687	26·211	8·824	729	27·000	9·000	771	27·767	9·169
688	26·229	8·828	730	27·018	9·004	772	27·785	9·173
689	26·249	8·832	731	27·037	9·008	773	27·803	9·177
690	26·268	8·836	732	27·055	9·012	774	27·821	9·181
691	26·287	8·841	733	27·074	9·016	775	27·839	9·185
692	26·306	8·845	734	27·092	9·020	776	27·857	9·189
693	26·325	8·849	735	27·111	9·023	777	27·875	9·193
694	26·344	8·853	736	27·129	9·029	778	27·893	9·197
695	26·363	8·858	737	27·148	9·033	779	27·910	9·201
696	26·382	8·862	738	27·166	9·037	780	27·928	9·205
697	26·401	8·866	739	27·184	9·041	781	27·946	9·209
698	26·419	8·870	740	27·203	9·045	782	27·964	9·213
699	26·439	8·875	741	27·221	9·049	783	27·982	9·217
700	26·457	8·879	742	27·239	9·053	784	28·000	9·221
701	26·476	8·883	743	27·258	9·057	785	28·017	9·225
702	26·495	8·887	744	27·276	9·061	786	28·036	9·229
703	26·514	8·892	745	27·295	9·065	787	28·053	9·233
704	26·533	8·896	746	27·313	9·069	788	28·071	9·237
705	26·552	8·900	747	27·331	9·073	789	28·089	9·240
706	26·571	8·904	748	27·349	9·077	790	28·107	9·244
707	26·589	8·908	749	27·368	9·081	791	28·125	9·248
708	26·608	8·913	750	27·386	9·086	792	28·142	9·252
709	26·627	8·917	751	27·404	9·089	793	28·160	9·256
710	26·644	8·921	752	27·423	9·094	794	28·178	9·260
711	26·664	8·925	753	27·441	9·098	795	28·196	9·264
712	26·683	8·929	754	27·459	9·102	796	28·213	9·268
713	26·702	8·934	755	27·477	9·106	797	28·231	9·271
714	26·721	8·938	756	27·495	9·109	798	28·249	9·275
715	26·739	8·942	757	27·514	9·114	799	28·266	9·279
716	26·758	8·946	758	27·532	9·118	800	28·284	9·283
717	26·777	8·950	759	27·549	9·122	801	28·302	9·287
718	26·795	8·954	760	27·568	9·126	802	28·319	9·291
719	26·814	8·959	761	27·586	9·129	803	28·337	9·295
720	26·833	8·963	762	27·604	9·134	804	28·355	9·299
721	26·851	8·967	763	27·622	9·138	805	28·372	9·302
722	26·870	8·971	764	27·640	9·142	806	28·390	9·306
723	26·889	8·975	765	27·659	9·146	807	28·408	9·310
724	26·907	8·979	766	27·677	9·149	808	28·425	9·314
725	26·926	8·983	767	27·695	9·154	809	28·443	9·318

TABLE 3 (*Continued*).

No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$
810	28·460	9·321	852	29·189	9·480	894	29·900	9·633
811	28·478	9·325	853	29·206	9·483	895	29·916	9·637
812	28·496	9·329	854	29·223	9·487	896	29·933	9·640
813	28·513	9·333	855	29·240	9·491	897	29·950	9·644
814	28·531	9·337	856	29·257	9·495	898	29·967	9·648
815	28·548	9·341	857	29·274	9·499	899	29·983	9·651
816	28·566	9·345	858	29·292	9·502	900	30·000	9·655
817	28·583	9·348	859	29·309	9·506	901	30·017	9·658
818	28·601	9·352	860	29·326	9·509	902	30·033	9·662
819	28·618	9·356	861	29·343	9·513	903	30·050	9·666
820	28·636	9·360	862	29·360	9·517	904	30·066	9·669
821	28·653	9·364	863	29·377	9·520	905	30·083	9·673
822	28·670	9·367	864	29·394	9·524	906	30·100	9·676
823	28·688	9·371	865	29·411	9·528	907	30·116	9·680
824	28·705	9·375	866	29·428	9·532	908	30·133	9·683
825	28·723	9·379	867	29·445	9·535	909	30·150	9·687
826	28·740	9·383	868	29·462	9·539	910	30·166	9·690
827	28·758	9·386	869	29·479	9·543	911	30·183	9·694
828	28·775	9·390	870	29·496	9·546	912	30·199	9·698
829	28·792	9·394	871	29·513	9·550	913	30·216	9·701
830	28·810	9·398	872	29·529	9·554	914	30·232	9·705
831	28·827	9·401	873	29·546	9·557	915	30·249	9·708
832	28·844	9·405	874	29·563	9·561	916	30·265	9·712
833	28·862	9·409	875	29·580	9·565	917	30·282	9·715
834	28·879	9·413	876	29·597	9·568	918	30·298	9·718
835	28·896	9·417	877	29·614	9·572	919	30·315	9·722
836	28·914	9·420	878	29·631	9·575	920	30·331	9·726
837	28·931	9·424	879	29·648	9·579	921	30·348	9·729
838	28·948	9·428	880	29·665	9·583	922	30·364	9·733
839	28·965	9·432	881	29·682	9·586	923	30·381	9·736
840	28·983	9·435	882	29·698	9·590	924	30·397	9·740
841	29·000	9·439	883	29·715	9·594	925	30·414	9·743
842	29·017	9·443	884	29·732	9·597	926	30·430	9·747
843	29·034	9·447	885	29·749	9·601	927	30·447	9·750
844	29·052	9·450	886	29·766	9·604	928	30·463	9·754
845	29·069	9·454	887	29·782	9·608	929	30·479	9·757
846	29·086	9·458	888	29·799	9·612	930	30·496	9·761
847	29·103	9·461	889	29·816	9·615	931	30·512	9·764
848	29·120	9·465	890	29·833	9·619	932	30·529	9·768
849	29·138	9·469	891	29·850	9·623	933	30·545	9·771
850	29·155	9·473	892	29·866	9·626	934	30·561	9·775
851	29·172	9·476	893	29·883	9·630	935	30·578	9·778

TABLE 3 (*Continued*).

No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$	No.	$\sqrt{N}$	$\sqrt[3]{N}$
936	30·594	9·781	958	30·951	9·858	980	31·305	9·933
937	30·610	9·785	959	30·968	9·861	981	31·321	9·936
938	30·627	9·789	960	30·984	9·865	982	31·337	9·940
939	30·643	9·792	961	31·000	9·868	983	31·353	9·943
940	30·659	9·796	962	31·016	9·872	984	31·369	9·946
941	30·676	9·799	963	31·032	9·875	985	31·385	9·950
942	30·692	9·803	964	31·048	9·878	986	31·401	9·953
943	30·708	9·806	965	31·064	9·881	987	31·416	9·956
944	80·724	9·810	966	31·080	9·885	988	31·432	9·960
945	30·741	9·813	967	31·097	9·889	989	31·448	9·963
946	30·757	9·817	968	31·113	9·892	990	31·464	9·966
947	30·773	9·820	969	31·129	9·895	991	31·480	9·970
948	30·790	9·823	970	31·145	9·899	992	31·496	9·973
949	30·806	9·827	971	31·161	9·902	993	31·512	9·977
950	30·822	9·830	972	31·177	9·906	994	31·528	9·980
951	30·838	9·834	973	31·193	9·909	995	31·544	9·983
952	30·854	9·837	974	31·209	9·912	996	31·559	9·987
953	30·871	9·841	975	31·225	9·916	997	31·575	9·990
954	30·887	9·844	976	31·241	9·919	998	31·591	9·993
955	30·903	9·848	977	31·257	9·923	999	31·607	9·997
956	30·919	9·851	978	31·273	9·926	1000	31·623	10·000
957	30·935	9·854	979	31·289	9·929	1001	31·638	10·000

TABLE 4.

VALUES OF $\sqrt{N}$ AND $\sqrt[3]{N}$ FOR N.

N advancing by ·01 to 1·00

Number.	$\sqrt{N}$	$\sqrt[3]{N}$	Number.	$\sqrt{N}$	$\sqrt[3]{N}$
0·01	0·10000	0·21544	0·51	0·71414	0·79896
0·02	0·14142	0·27144	0·52	0·72111	0·80415
0·03	0·17321	0·31072	0·53	0·72801	0·80927
0·04	0·20000	0·34200	0·54	0·73485	0·81433
0·05	0·22361	0·36840	0·55	0·74162	0·81932
0·06	0·24495	0·39149	0·56	0·74833	0·82426
0·07	0·26458	0·41213	0·57	0·75498	0·82913
0·08	0·28284	0·43089	0·58	0·76158	0·83396
0·09	0·30000	0·44814	0·59	0·76811	0·83872
0·10	0·31623	0·46416	0·60	0·77460	0·84343
0·11	0·33166	0·47914	0·61	0·78102	0·84809
0·12	0·34641	0·49324	0·62	0·78740	0·85270
0·13	0·36056	0·50658	0·63	0·79373	0·85726
0·14	0·37417	0·51925	0·64	0·80000	0·86177
0·15	0·38730	0·53133	0·65	0·80623	0·86624
0·16	0·40000	0·54288	0·66	0·81240	0·87066
0·17	0·41231	0·55397	0·67	0·81844	0·87503
0·18	0·42426	0·56462	0·68	0·82462	0·87937
0·19	0·42589	0·57489	0·69	0·83066	0·88366
0·20	0·44721	0·58480	0·70	0·83666	0·88790
0·21	0·45826	0·59439	0·71	0·84261	0·89211
0·22	0·46904	0·60368	0·72	0·84853	0·89628
0·23	0·47958	0·61269	0·73	0·85440	0·90041
0·24	0·48990	0·62145	0·74	0·86023	0·90450
0·25	0·50000	0·62996	0·75	0·86603	0·90856
0·26	0·50990	0·63825	0·76	0·87178	0·91258
0·27	0·51962	0·64633	0·77	0·87750	0·91657
0·28	0·52915	0·65421	0·78	0·88318	0·92052
0·29	0·53852	0·66191	0·79	0·88882	0·92443
0·30	0·54772	0·66943	0·80	0·89443	0·92832
0·31	0·55678	0·67679	0·81	0·90000	0·93217
0·32	0·56569	0·68399	0·82	0·90554	0·93599
0·33	0·57446	0·69104	0·83	0·91104	0·93978
0·34	0·58310	0·69795	0·84	0·91652	0·94354
0·35	0·59161	0·70473	0·85	0·92195	0·94727
0·36	0·60000	0·71138	0·86	0·92736	0·95097
0·37	0·60828	0·71791	0·87	0·93274	0·95464
0·38	0·61644	0·72432	0·88	0·93808	0·95828
0·39	0·62450	0·73061	0·89	0·94340	0·96190
0·40	0·63246	0·73681	0·90	0·94868	0·96549
0·41	0·64031	0·74290	0·91	0·95394	0·96905
0·42	0·64807	0·74889	0·92	0·95917	0·97259
0·43	0·65574	0·75478	0·93	0·96437	0·97610
0·44	0·66332	0·76059	0·94	0·96954	0·97959
0·45	0·67082	0·76631	0·95	0·97468	0·98305
0·46	0·67823	0·77194	0·96	0·97980	0·98648
0·47	0·68557	0·77750	0·97	0·98489	0·98990
0·48	0·69282	0·78297	0·98	0·98995	0·99329
0·49	0·70000	0·78837	0·99	0·99499	0·99666
0·50	0·70711	0·79370	1·00	1·00000	1·00000

TABLE 5.

APPROXIMATE FOURTH AND FIFTH ROOTS.

No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$
1	1·0000	1·000	40	2·5132	2·091	79	2·9808	2·396
2	1·2071	1·149	41	2·5286	2·101			
3	1·3032	1·246	42	2·5438	2·112	80	2·9904	2·402
4	1·4142	1·319	43	2·5589	2·122	81	3·0000	2·408
5	1·4892	1·379	44	2·5737	2·132	82	3·0089	2·414
6	1·5571	1·431	45	2·5885	2·141	83	3·0179	2·419
7	1·6194	1·476	46	2·6030	2·150	84	3·0268	2·425
8	1·6775	1·516	47	2·6174	2·160	85	3·0356	2·432
9	1·7321	1·552	48	2·6317	2·169	86	3·0444	2·437
			49	2·6458	2·178	87	3·0531	2·443
10	1·7755	1·585				88	3·0618	2·448
11	1·8169	1·615	50	2·6587	2·187	89	3·0704	2·454
12	1·8564	1·644	51	2·6716	2·196			
13	1·8943	1·670	52	2·6843	2·204	90	3·0790	2·460
14	1·9308	1·695	53	2·6969	2·212	91	3·0875	2·465
15	1·9659	1·719	54	2·7094	2·221	92	3·0960	2·470
16	2·0000	1·741	55	2·7217	2·229	93	3·1044	2·476
17	2·0290	1·762	56	2·7340	2·237	94	3·1128	2·480
18	2·0572	1·782	57	2·7461	2·245	95	3·1212	2·486
19	2·0847	1·802	58	2·7582	2·253	96	3·1295	2·491
			59	2·7701	2·260	97	3·1377	2·497
20	2·1114	1·821				98	3·1459	2·501
21	2·1375	1·838	60	2·7820	2·268	99	3·1541	2·507
22	2·1630	1·856	61	2·7937	2·276			
23	2·1878	1·872	62	2·8053	2·283	100	3·1623	2·512
24	2·2122	1·888	63	2·8169	2·290	101	3·1706	2·518
25	2·2361	1·904	64	2·8284	2·297	102	3·1776	2·522
26	2·2572	1·919	65	2·8390	2·304	103	3·1860	2·526
27	2·2779	1·933	66	2·8496	2·312	104	3·1928	2·531
28	2·2983	1·947	67	2·8602	2·318	105	3·2010	2·536
29	2·3193	1·961	68	2·8706	2·326	106	3·2079	2·541
			69	2·8810	2·332	107	3·2160	2·546
30	2·3379	1·974				108	3·2228	2·551
31	2·3572	1·987	70	2·8913	2·339	109	3·231	2·555
32	2·3762	2·002	71	2·9015	2·345			
33	2·3950	2·012	72	2·9116	2·352	110	3·2376	2·560
34	2·4034	2·024	73	2·9217	2·358	111	3·2460	2·565
35	2·4315	2·036	74	2·9317	2·364	112	3·2522	2·569
36	2·4495	2·047	75	2·9417	2·371	113	3·2600	2·574
37	2·4657	2·059	76	2·9515	2·378	114	3·2667	2·578
38	2·4817	2·070	77	2·9613	2·384	115	3·2750	2·583
39	2·4975	2·081	78	2·9711	2·390	116	3·2811	2·587

TABLE 5. (*Continued*).

No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$
117	3·2930	2·592	155	3·5290	2·742	220	3·8509	2·941
118	3·2954	2·596	156	3·5334	2·746	222	3·8598	2·946
119	3·3030	2·601	157	3·5400	2·749	224	3·8686	2·951
			158	3·5447	2·752	226	3·8772	2·956
120	3·3095	2·605	159	3·5510	2·756	228	3·8856	2·962
121	3·3160	2·609						
122	3·3233	2·613	160	3·5559	2·759	230	3·8940	2·967
123	3·3300	2·618	162	3·5671	2·767	232	3·9024	2·973
124	3·3365	2·622	164	3·5782	2·773	234	3·9107	2·977
125	3·3440	2·626	166	3·5892	2·780	236	3·9190	2·983
126	3·3497	2·630	168	3·6001	2·786	238	3·9272	2·987
127	3·3570	2·634						
128	3·3628	2·639	170	3·6108	2·793	240	3·9354	2·992
129	3·3700	2·643	172	3·6212	2·800	242	3·9436	2·997
			174	3·6315	2·806	244	3·9518	3·002
130	3·3758	2·646	176	3·6418	2·813	246	3·9599	3·007
131	3·3830	2·651	178	3·6521	2·818	248	3·9639	3·012
132	3·3887	2·656						
133	3·3960	2·659	180	3·6622	2·825	250	3·9720	3·017
134	3·4020	2·663	182	3·6723	2·831	252	3·9840	3·022
135	3·4080	2·667	184	3·6824	2·837	254	3·9920	3·027
136	3·4142	2·672	186	3·6924	2·844	256	4·0000	3·031
137	3·4210	2·676	188	3·7024	2·849	258	4·0076	3·036
138	3·4268	2·678						
139	3·4340	2·682	190	3·7123	2·856	260	4·0153	3·041
			192	3·7221	2·862	262	4·0229	3·046
140	3·4393	2·686	194	3·7319	2·867	264	4·0305	3·050
141	3·4460	2·691	196	3·7417	2·874	266	4·0380	3·055
142	3·4477	2·695	198	3·7510	2·879	268	4·0456	3·059
143	3·4580	2·698						
144	3·4641	2·702	200	3·7603	2·885	270	4·0531	3·064
145	3·4700	2·706	202	3·7696	2·892	272	4·0605	3·068
146	3·4758	2·710	204	3·7788	2·897	274	4·0679	3·073
147	3·4820	2·713	206	3·7878	2·902	276	4·0754	3·077
148	3·4875	2·716	208	3·7971	2·908	278	4·0828	3·082
149	3·4940	2·720						
			210	3·8062	2·914	280	4·0902	3·086
150	3·4991	2·724	212	3·8152	2·919	282	4·0974	3·090
151	3·5050	2·728	214	3·8242	2·925	284	4·1048	3·095
152	3·5106	2·731	216	3·8332	2·930	286	4·1121	3·099
153	3·5170	2·735	218	3·8421	2·936	288	4·1194	3·103
154	3·5220	2·738						

TABLE 5 (*Continued*).

No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$
290	4·1265	3·109	360	4·3558	3·245	430	4·5534	3·362
292	4·1336	3·113	362	4·3618	3·248	432	4·5588	3·366
294	4·1405	3·117	364	4·3678	3·253	434	4·5641	3·369
296	4·1475	3·121	366	4·3737	3·256	436	4·5694	3·372
298	4·1545	3·125	368	4·3796	3·260	438	4·5746	3·375
300	4·1613	3·129	370	4·3855	3·263	440	4·5799	3·378
302	4·1682	3·133	372	4·3913	3·267	442	4·5851	3·382
304	4·1752	3·137	374	4·3972	3·271	444	4·5902	3·385
306	4·1820	3·142	376	4·4031	3·273	446	4·5954	3·388
308	4·1887	3·146	378	4·4089	3·277	448	4·6004	3·390
310	4·1956	3·149	380	4·4147	3·281	450	4·6055	3·393
312	4·2023	3·154	382	4·4205	3·284	452	4·6106	3·396
314	4·2091	3·157	384	4·4263	3·288	454	4·6156	3·399
316	4·2158	3·161	386	4·4321	3·291	456	4·6207	3·402
318	4·2225	3·165	388	4·4379	3·295	458	4·6258	3·406
320	4·2292	3·170	390	4·4435	3·298	460	4·6307	3·409
322	4·2359	3·174	392	4·4493	3·301	462	4·6358	3·411
324	4·2426	3·178	394	4·4550	3·305	464	4·6409	3·414
326	4·2489	3·181	396	4·4606	3·308	466	4·6458	3·418
328	4·2555	3·185	398	4·4663	3·311	468	4·6508	3·420
330	4·2619	3·189	400	4·4721	3·314	470	4·6557	3·423
332	4·2683	3·193	402	4·4755	3·317	472	4·6607	3·426
334	4·2746	3·197	404	4·4830	3·321	474	4·6657	3·430
336	4·2809	3·201	406	4·4885	3·324	476	4·6706	3·432
338	4·2873	3·204	408	4·4940	3·328	478	4·6805	3·435
340	4·2936	3·208	410	4·4995	3·332	480	4·6854	3·438
342	4·2999	3·212	412	4·5050	3·334	482	4·6904	3·441
344	4·3062	3·216	414	4·5104	3·339	484	4·6934	3·443
346	4·3124	3·220	416	4·5158	3·341	486	4·6959	3·445
348	4·3187	3·223	418	4·5212	3·344	488	4·6999	3·449
350	4·3249	3·227	420	4·5266	3·347	490	4·7047	3·451
352	4·3312	3·231	422	4·5321	3·351	492	4·7094	3·454
354	4·3373	3·234	424	4·5374	3·354	494	4·7142	3·457
356	4·3435	3·238	426	4·5427	3·357	496	4·7189	3·460
358	4·3497	3·242	428	4·5481	3·359	498	4·7237	3·462

TABLE 5 (*Continued*).

No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$
500	4·7284	3·465	570	4·886	3·558	640	5·029	3·641
502	4·7330	3·468	572	4·890	3·561	642	5·034	3·643
504	4·7380	3·471	574	4·899	3·564	644	5·038	3·645
506	4·7430	3·474	576	4·903	3·566	646	5·042	3·649
508	4·7470	3·477	578	4·907	3·568	648	5·046	3·650
510	4·752	3·479	580	4·907	3·570	650	5·049	3·652
512	4·756	3·482	582	4·912	3·572	652	5·054	3·655
514	4·761	3·485	584	4·916	3·575	654	5·057	3·657
516	4·766	3·488	586	4·920	3·577	656	5·061	3·659
518	4·770	3·491	588	4·924	3·580	658	5·064	3·661
520	4·775	3·493	590	4·928	3·582	660	5·068	3·664
522	4·780	3·496	592	4·933	3·585	662	5·072	3·666
524	4·785	3·498	594	4·937	3·587	664	5·076	3·668
526	4·789	3·501	596	4·941	3·590	666	5·080	3·670
528	4·794	3·504	598	4·945	3·592	668	5·083	3·672
530	4·798	3·507	600	4·949	3·594	670	5·087	3·675
532	4·802	3·509	602	4·953	3·597	672	5·091	3·677
534	4·807	3·512	604	4·957	3·599	674	5·095	3·679
536	4·811	3·514	606	4·961	3·601	676	5·099	3·681
538	4·816	3·517	608	4·965	3·604	678	5·103	3·683
540	4·820	3·520	610	4·969	3·606	680	5·106	3·686
542	4·825	3·522	612	4·975	3·609	682	5·110	3·688
544	4·829	3·525	614	4·978	3·611	684	5·114	3·690
546	4·833	3·527	616	4·981	3·613	686	5·118	3·692
548	4·838	3·530	618	4·986	3·615	688	5·122	3·694
550	4·842	3·533	620	4·989	3·618	690	5·125	3·696
552	4·847	3·535	622	4·994	3·620	692	5·129	3·699
554	4·851	3·539	624	4·997	3·623	694	5·133	3·701
556	4·856	3·541	626	5·001	3·625	696	5·136	3·703
558	4·860	3·543	628	5·005	3·627	698	5·140	3·705
560	4·864	3·545	630	5·009	3·629	700	5·143	3·707
562	4·869	3·548	632	5·014	3·632	705	5·152	3·712
564	4·873	3·550	634	5·018	3·634			
566	4·878	3·553	636	5·022	3·636	710	5·161	3·717
568	4·882	3·555	638	5·026	3·639	715	5·170	3·722

TABLE 5 (*Continued*).

No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$	No.	$\sqrt[4]{N}$	$\sqrt[5]{N}$
720	5·179	3·725	815	5·342	3·821	910	5·491	3·907
725	5·189	3·733				915	5·499	3·911
			820	5·351	3·826			
730	5·197	3·739	825	5·359	3·831	920	5·507	3·916
735	5·206	3·744				925	5·514	3·920
			830	5·367	3·835			
740	5·215	3·748	835	5·375	3·840	930	5·522	3·924
745	5·224	3·753				935	5·529	3·928
			840	5·383	3·845			
750	5·234	3·758	845	5·391	3·850	940	5·536	3·932
755	5·241	3·763				945	5·544	3·937
			850	5·399	3·854			
760	5·250	3·768	855	5·407	3·859	950	5·551	3·940
765	5·259	3·773				955	5·559	3·944
			860	5·415	3·863			
770	5·267	3·778	865	5·423	3·868	960	5·566	3·949
775	5·276	3·783				965	5·573	3·953
			870	5·430	3·872			
780	5·284	3·788	875	5·438	3·879	970	5·580	3·957
785	5·293	3·793				975	5·587	3·962
			880	5·446	3·881			
790	5·301	3·797	885	5·454	3·886	980	5·594	3·965
795	5·309	3·803				985	5·602	3·969
			890	5·461	3·890			
800	5·318	3·807	895	5·469	3·894	990	5·609	3·973
805	5·326	3·812				995	5·616	3·977
			900	5·477	3·897			
810	5·334	3·817	905	5·484	3·902	1000	5·623	3·981

TABLE 6.
SQUARE ROOTS OF THE FIFTH POWERS.

No.	$\sqrt{N^5}$	No.	$\sqrt{N^5}$	No.	$\sqrt{N^5}$	No.	$\sqrt{N^5}$
·25	·031	8·75	226·48	22·	2270·2	36·	7776·
·5	·177			22·5	2401·4	36·5	8049·
·75	·485	9·	243·				
		9·25	260·23	23·	2537·	37·	8327
1·	1·	9·5	278·17	23·5	2677·1	37·5	8611
1·25	1·747	9·75	296·83				
1·5	2·756			24·	2821·8	38·	8901
1·75	4·051	10·	316·23	24·5	2971·1	38·5	9197
		10·5	357·2				
2·	5·657			25·	3125·	39·	9498
2·25	7·594	11·	401·3	25·5	3283·6	39·5	9806
2·5	9·882	11·5	448·5				
2·75	12·541			26·	3446·9	40	10119
		12·	498·8	26·5	3615·1	41	10764
3·	15·588	12·5	552·4			42	11432
3·25	19·042			27·	3788·	43	12125
3·5	22·918	13·	609·3	27·5	3965·8	44	12842
3·75	27·232	13·5	669·6			45	13584
				28·	4148·5	46	14351
4·	32·	14·	733·4	28·5	4336·2	47	15144
4·25	37·24	14·5	800·6			48	15963
4·5	42·96			29·	4528·9	49	16807
4·75	49·17	15·	871·4	29·5	4726·7		
		15·5	945·9			50	17678
5·	55·9			30·	4929·5	51	18575
5·25	63·15	16·	1024·	30·5	5138·	52	19499
5·5	70·94	16·5	1105·9			53	20450
5·75	79·28			31·	5351·	54	21428
		17·	1191·6	31·5	5569·	55	22434
6·	88·18	17·5	1281·1			56	23468
6·25	97·66			32·	5793·	57	24529
6·5	107·72	18·	1374·6	32·5	6022·	58	25620
6·75	118·38	18·5	1472·1			59	26738
				33·	6256·		
7·	129·64	19·	1573·6	33·5	6496·	60	27886
7·25	141·53	19·5	1679·1			61	29062
7·5	154·05			34·	6741·	62	30268
7·75	167·21	20·	1788·9	34·5	6991·	63	31503
		20·5	1902·8			64	32768
8·	181·02			35·	7247·	65	34063
8·25	195·5	21·	2020·9	35·5	7509·	66	35388
8·5	210·64	21·5	2143·4			67	36744

TABLE 6 (*Continued*).

No.	$\sqrt{N^5}$	No.	$\sqrt{N^5}$	No.	$\sqrt{N^5}$	No.	$\sqrt{N^5}$
68	38131	89	74727	110	126906	131	196416
69	39548			111	129810	132	200186
		90	76843	112	132753	133	203999
70	40996	91	78996	113	135736	134	207856
71	42476	92	81184	114	138759	135	211755
72	43988	93	83408	115	141822	136	215699
73	45531	94	85668	116	144925	137	219685
74	47106	95	87965	117	148069	138	223716
75	48714	96	90298	118	151253	139	227791
76	50354	97	92668	119	154154		
77	52027	98	95075			140	231910
78	53732	99	97519	120	157744	141	236074
79	55471			121	161051	142	240282
		100	100000	122	164399	143	244534
80	57243	101	102519	123	167789	144	248832
81	59049	102	105075	124	171220	145	253174
82	60888	103	107669	125	174693	146	257562
83	62762	104	110302	126	178208	147	261995
84	64669	105	112973	127	181765	148	266474
85	66611	106	115682	128	185364	149	270998
86	68588	107	118429	129	189005		
87	70509	108	121216			150	275567
88	72645	109	124041	130	192690		

TABLE 7.

Value of $\frac{1}{N}$ and Log $\frac{1}{N}$ for N

N advancing by 1 to 250

N	$\frac{1}{N}$	Log $\frac{1}{N}$	N	$\frac{1}{N}$	Log $\frac{1}{N}$
1	1·00000	$\bar{1}$·00000	41	·02439	$\bar{2}$·38722
2	·50000	$\bar{1}$·69897	42	·02381	$\bar{2}$·37675
3	·33333	$\bar{1}$ 52288	43	·02326	$\bar{2}$·36653
4	·25000	$\bar{1}$ 39794	44	·02273	$\bar{2}$·35655
5	·20000	$\bar{1}$·30103	45	·02222	$\bar{2}$·34679
6	·16667	$\bar{1}$·22185	46	·02174	$\bar{2}$·33724
7	·14286	$\bar{1}$·15490	47	·02128	$\bar{2}$·32790
8	·12500	$\bar{1}$·09691	48	·02083	$\bar{2}$·31876
9	·11111	$\bar{1}$·04576	49	·02041	$\bar{2}$·30980
10	·10000	$\bar{1}$·00000	50	·02000	$\bar{2}$·30103
11	·09091	$\bar{2}$·95861	51	·01961	$\bar{2}$·29243
12	·08333	$\bar{2}$ 92082	52	·01923	$\bar{2}$·28400
13	·07692	$\bar{2}$·88696	53	·01887	$\bar{2}$·27572
14	·07143	$\bar{2}$·85387	54	·01852	$\bar{2}$·26761
15	·06667	$\bar{2}$·82391	55	·01818	$\bar{2}$·25964
16	·06250	$\bar{2}$ 79588	56	·01786	$\bar{2}$·25181
17	·05882	$\bar{2}$ 76955	57	·01754	$\bar{2}$·24413
18	·05556	$\bar{2}$ 74473	58	·01724	$\bar{2}$·23657
19	·05263	$\bar{2}$·72125	59	·01695	$\bar{2}$·22915
20	·05000	$\bar{2}$ 69897	60	·01667	$\bar{2}$ 22185
21	·04762	$\bar{2}$ 67778	61	·01639	$\bar{2}$·21447
22	·04545	$\bar{2}$ 65758	62	·01613	$\bar{2}$·20761
23	·04348	$\bar{2}$·63827	63	·01587	$\bar{2}$ 20066
24	·04167	$\bar{2}$ 61979	64	·01563	$\bar{2}$·19382
25	·04000	$\bar{2}$ 60206	65	·01538	$\bar{2}$·18709
26	·03846	$\bar{2}$·58503	66	·01515	$\bar{2}$ 18046
27	·03704	$\bar{2}$ 56864	67	·01493	$\bar{2}$·17393
28	·03571	$\bar{2}$ 55824	68	·01471	$\bar{2}$·16749
29	·03448	$\bar{2}$ 53760	69	·01449	$\bar{2}$ 16115
30	·03333	$\bar{2}$ 52288	70	·01429	$\bar{2}$ 15490
31	·03226	$\bar{2}$ 50864	71	·01408	$\bar{2}$·14874
32	·03125	$\bar{2}$ 49485	72	·01389	$\bar{2}$·14267
33	·03030	$\bar{2}$ 48149	73	·01370	$\bar{2}$·13668
34	·02941	$\bar{2}$ 46852	74	·01351	$\bar{2}$·13077
35	·02857	$\bar{2}$ 45593	75	·01333	$\bar{2}$·12494
36	·02778	$\bar{2}$ 44370	76	·01316	$\bar{2}$·11919
37	·02703	$\bar{2}$·43180	77	·01299	$\bar{2}$·11351
38	·02632	$\bar{2}$·42522	78	·01282	$\bar{2}$·10791
39	·02564	$\bar{2}$·40894	79	·01266	$\bar{2}$·10237
40	·02500	$\bar{2}$·39794	80	·01250	$\bar{2}$ 09691

TABLE 7 (*Continued*).

N advancing by 1 to 250

N	$\frac{1}{N}$	Log $\frac{1}{N}$	N	$\frac{1}{N}$	Log $\frac{1}{N}$
81	·01235	$\bar{2}$·09151	125	·00800	$\bar{3}$·90309
82	·01220	$\bar{2}$·08619	126	·00794	$\bar{3}$·89963
83	·01205	$\bar{2}$·08092	127	·00787	$\bar{3}$·89620
84	·01190	$\bar{2}$·07572	128	·00781	$\bar{3}$·89279
85	·01176	$\bar{2}$·07058	129	·00775	$\bar{3}$·88941
86	·01163	$\bar{2}$·06550	130	·00769	$\bar{3}$·88606
87	·01149	$\bar{2}$·06048	131	·00763	$\bar{3}$·88273
88	·01136	$\bar{2}$·05552	132	·00758	$\bar{3}$·87943
89	·01124	$\bar{2}$·05061	133	·00752	$\bar{3}$·87615
90	·01111	$\bar{2}$·04576	134	·00746	$\bar{3}$·87290
91	·01099	$\bar{2}$·04096	135	·00741	$\bar{3}$·86967
92	·01087	$\bar{2}$·03621	136	·00735	$\bar{3}$·86646
93	·01075	$\bar{2}$·03152	137	·00730	$\bar{3}$·86328
94	·01064	$\bar{2}$·02687	138	·00725	$\bar{3}$·86012
95	·01053	$\bar{2}$·02228	139	·00719	$\bar{3}$·85699
96	·01042	$\bar{2}$·01773	140	·00714	$\bar{3}$·85387
97	·01031	$\bar{2}$·01323	141	·00709	$\bar{3}$·85078
98	·01020	$\bar{2}$·00877	142	·00704	$\bar{3}$·84771
99	·01010	$\bar{2}$·00436	143	·00699	$\bar{3}$·84466
100	·01000	$\bar{2}$·00000	144	·00694	$\bar{3}$·84164
101	·00990	$\bar{3}$·99568	145	·00690	$\bar{3}$·83863
102	·00980	$\bar{3}$·99140	146	·00685	$\bar{3}$·83565
103	·00971	$\bar{3}$·98716	147	·00680	$\bar{3}$·83268
104	·00962	$\bar{3}$·98297	148	·00676	$\bar{3}$·82970
105	·00952	$\bar{3}$·97881	149	·00671	$\bar{3}$·82681
106	·00943	$\bar{3}$·97469	150	·00667	$\bar{3}$·82391
107	·00935	$\bar{3}$·97062	151	·00662	$\bar{3}$·82102
108	·00926	$\bar{3}$·96658	152	·00658	$\bar{3}$·81816
109	·00917	$\bar{3}$·96257	153	·00654	$\bar{3}$·81531
110	·00909	$\bar{3}$·95861	154	·00649	$\bar{3}$·81248
111	·00901	$\bar{3}$·95468	155	·00645	$\bar{3}$·80967
112	·00893	$\bar{3}$·95078	156	·00641	$\bar{3}$·80688
113	·00885	$\bar{3}$·94692	157	·00637	$\bar{3}$·80410
114	·00877	$\bar{3}$·94310	158	·00633	$\bar{3}$·80134
115	·00870	$\bar{3}$·93930	159	·00629	$\bar{3}$·79860
116	·00862	$\bar{3}$·93554	160	·00625	$\bar{3}$·79588
117	·00855	$\bar{3}$·93181	161	·00621	$\bar{3}$·79317
118	·00847	$\bar{3}$·92812	162	·00617	$\bar{3}$·79048
119	·00840	$\bar{3}$·92445	163	·00613	$\bar{3}$·78781
120	·00833	$\bar{3}$·92082	164	·00610	$\bar{3}$·78516
121	·00826	$\bar{3}$·91721	165	·00606	$\bar{3}$·78252
122	·00820	$\bar{3}$·91364	166	·00602	$\bar{3}$·78089
123	·00813	$\bar{3}$·91009	167	·00599	$\bar{3}$·77728
124	·00806	$\bar{3}$·90658	168	·00595	$\bar{3}$·77469

TABLE 7 (*Continued*).

N advancing by 1 to 250

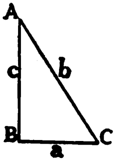
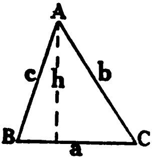
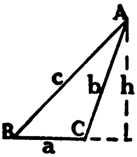
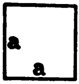
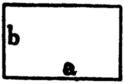
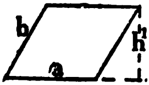
N	$\frac{1}{N}$	Log $\frac{1}{N}$	N	$\frac{1}{N}$	Log $\frac{1}{N}$
169	·00592	$\bar{3}$·77211	210	·00476	$\bar{3}$·67778
170	·00588	$\bar{3}$·76955	211	·00474	$\bar{3}$·67572
171	·00585	$\bar{3}$·76700	212	·00472	$\bar{3}$·67366
172	·00581	$\bar{3}$·76447	213	·00469	$\bar{3}$ 67162
173	·00578	$\bar{3}$·76195	214	·00467	$\bar{3}$·66959
174	·00575	$\bar{3}$·75945	215	·00465	$\bar{3}$·66756
175	·00571	$\bar{3}$·75696	216	·00463	$\bar{3}$·66555
176	·00568	$\bar{3}$·75449	217	·00461	$\bar{3}$·66354
177	·00565	$\bar{3}$·75203	218	·00459	$\bar{3}$ 66154
178	·00562	$\bar{3}$·74958	219	·00457	$\bar{3}$·65956
179	·00559	$\bar{3}$·74715	220	·00455	$\bar{3}$ 65758
180	·00556	$\bar{3}$·74473	221	·00452	$\bar{3}$ 65561
181	·00552	$\bar{3}$·74232	222	·00450	$\bar{3}$ 65365
182	·00549	$\bar{3}$·73993	223	·00448	$\bar{3}$ 65170
183	·00546	$\bar{3}$·73755	224	·00446	$\bar{3}$·64975
184	·00543	$\bar{3}$·73518	225	·00444	$\bar{3}$ 64782
185	·00541	$\bar{3}$·73283	226	·00442	$\bar{3}$·64589
186	·00538	$\bar{3}$·73049	227	·00441	$\bar{3}$ 64397
187	·00535	$\bar{3}$·72816	228	·00439	$\bar{3}$·64207
188	·00532	$\bar{3}$·72584	229	·00437	$\bar{3}$ 64016
189	·00529	$\bar{3}$·72354	230	·00435	$\bar{3}$ 63827
190	·00526	$\bar{3}$·72125	231	·00433	$\bar{3}$·63639
191	·00524	$\bar{3}$·71897	232	·00431	$\bar{3}$·63451
192	·00521	$\bar{3}$·71670	233	·00429	$\bar{3}$·63264
193	·00518	$\bar{3}$·71444	234	·00427	$\bar{3}$ 63078
194	·00515	$\bar{3}$·71220	235	·00426	$\bar{3}$·62893
195	·00513	$\bar{3}$·70997	236	·00424	$\bar{3}$·62709
196	·00510	$\bar{3}$·70774	237	·00422	$\bar{3}$·62525
197	·00508	$\bar{3}$·70553	238	·00420	$\bar{3}$·62342
198	·00505	$\bar{3}$·70333	239	·00418	$\bar{3}$·62160
199	·00503	$\bar{3}$·70115	240	·00417	$\bar{3}$·61979
200	·00500	$\bar{3}$·69897	241	·00415	$\bar{3}$ 61798
201	·00498	$\bar{3}$·69680	242	·00413	$\bar{3}$·61618
202	·00495	$\bar{3}$·69465	243	·00412	$\bar{3}$·61439
203	·00493	$\bar{3}$·69250	244	·00410	$\bar{3}$ 61261
204	·00490	$\bar{3}$·69037	245	·00408	$\bar{3}$·61083
205	·00488	$\bar{3}$·68825	246	·00407	$\bar{3}$·60906
206	·00485	$\bar{3}$·68613	247	·00405	$\bar{3}$·60730
207	·00483	$\bar{3}$·68403	248	·00403	$\bar{3}$·60556
208	·00481	$\bar{3}$·68194	249	·00402	$\bar{3}$·60380
209	·00478	$\bar{3}$·67985	250	·00400	$\bar{3}$ 60206

TABLE 8.

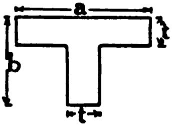
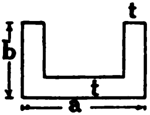
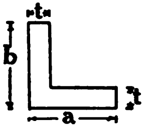
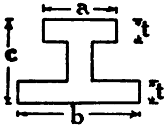
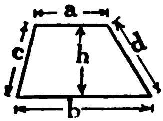
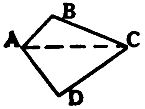
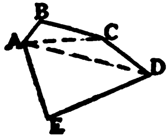
Functions of π, g, e.

Expression	Numerical Value	Logarithm.	Expression	Numerical Value	Logarithm.
π	3·1416	0·4971	g	32·189	1·5077
2π	6·2831	0·7882	$2g$	64·378	1·8087
$\frac{\pi}{2}$	1·5708	0·1961	g^2	1036·5	3·0154
π^2	9·8696	0·9943	$\frac{1}{g}$	·0311	$\bar{2}$·4923
$4\pi^2$	39·4784	1·5964	$\frac{1}{g^2}$	·0009652	$\bar{4}$·9846
$\frac{\pi^2}{4}$	2·4674	0·3922	$\sqrt{\frac{1}{g}}$	·1762	$\bar{1}$·2461
$\sqrt{\pi}$	1·7724	0·2485	$\sqrt{2g}$	8·031	0·9043
$\pi\sqrt{\pi}$	5·5683	0·7457	$\frac{1}{\sqrt{2g}}$	·12465	$\bar{1}$·0957
$\sqrt{2\pi}$	2·5066	0·3991	e	2·7183	0·4343
$\sqrt{\frac{\pi}{2}}$	1·2533	0·0981	$\frac{1}{e}$	0·3679	$\bar{1}$·5657
$\sqrt{\frac{2}{\pi}}$	0·7979	$\bar{1}$·9019	e^2	7·3890	0·8686
$\sqrt[3]{\pi}$	1·4645	0·1657	$\frac{1}{e^2}$	0·1353	$\bar{1}$·1314
$\frac{\pi}{180}$	·017453	$\bar{2}$·24187	$\sqrt{e}$	1·6487	0·2171
$\frac{180}{\pi}$	57·29577	1·75812	$\sqrt[3]{e}$	1·3956	0·1448

MENSURATION OF PLANE SURFACES.

FORM.	NAME.	PERIMETER.	AREA.
	Right-angled Triangle	Perimeter $= a + b + c$ *Note* $s = \frac{a+b+c}{2}$ $a^2 + c^2 = b^2$	Area $S = \frac{a \times c}{2}$
	Acute-angled Triangle	Perimeter $= a + b + c$ $s = \frac{a+b+c}{2}$	Area $S = \frac{a \times h}{2}$
	Obtuse-angled Triangle		
	Square	Perimeter $= 4a$	Area $= a^2$
	Rectangle	Perimeter $= 2a + 2b$	Area $= a \times b$
	Parallelogram	Perimeter $= 2a + 2b$	Area $= a \times h$

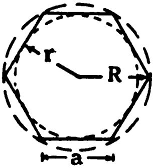
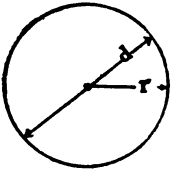
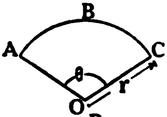
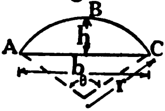
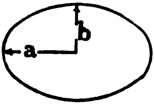
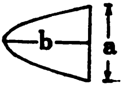
MENSURATION OF

FORM.	NAME.
	Tee Section
	Channel or U Section
	Angle or L Section
	I Section
	Trapezium
	Quadrilateral
	Irregular Polygon

PLANE SURFACES

PERIMETER.	AREA.
Perimeter $= 2a + 2b$	Area $= at + bt - t^2$
Perimeter $= 2(a + 2b - t)$	Area $= at + 2bt - 2t^2$
Perimeter $= 2a + 2b$	Area $= at + bt - t^2$
Perimeter $= 2a + 2b + 2c - 2t$	Area $= at + bt + ct - t^2$
Perimeter $= a + b + c + d$	Area $= \dfrac{a + b}{2} \times h$
Perimeter $= \left\{ \begin{array}{l} AB + BC + CD \\ + DA \end{array} \right.$	Area $= \left\{ \begin{array}{l} \text{Area triangle} \\ \text{ABC + area} \\ \text{triangle} \\ \text{ACD} \end{array} \right.$
Perimeter $= \left\{ \begin{array}{l} AB + BC + CD \\ + DE + EA \end{array} \right.$	Area $= \left\{ \begin{array}{l} \text{Sum of areas} \\ \text{of triangles} \\ \text{ABC, ACD,} \\ \text{ADE} \end{array} \right.$

MENSURATION OF

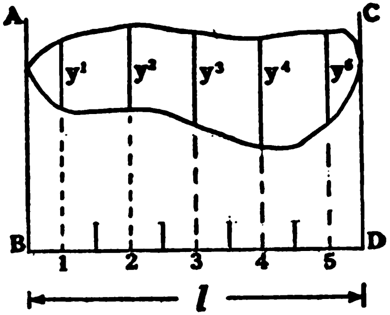
FORM.	NAME.	PERIMETER.
	Regular Polygon of n sides When $n =$ 5 Polygon is known as ,, $n =$ 6 ,, ,, ,, $n =$ 8 ,, ,, ,, $n =$ 10 ,, ,,	Perimeter $= n \times a$ · *Note* $r =$ radius of R= ,,
	Circle	Perimeter
	Segment of Circle	Perimeter
	Sector of Circle	Perimeter
	Ellipse	$4\sqrt{a^2 + 1{\cdot}4674b^2}$
	Parabola	Perimeter

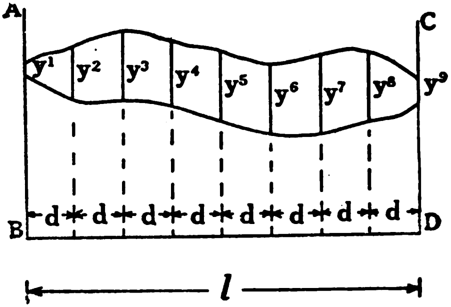
PLANE SURFACES

Perimeter (*cont.*)—	Area.
	Area $= \frac{1}{2} \times a \times r \times n$
inscribed circle circumscribed circle	
Pentagon $r = a \times \cdot 688$; $R = a \times \cdot 851$ Hexagon $r = a \times \cdot 866$; $R = a \times 1$ Octagon $r = a \times 1{\cdot}217$; $R = a \times 1{\cdot}309$ Decagon $r = a \times 1{\cdot}538$; $R = a \times 1{\cdot}618$	
$= 2\pi r$ $= 6{\cdot}2832 \times r$ $= \pi.d$ $= 3{\cdot}1416.d$	Area $= \pi r^2$ $= 3{\cdot}1416\ r^2$ or $\frac{22}{7} r^2$ $= \pi\ 4{:}d^2$ $= \cdot 7854 d^2$
$=$ arc ABC $+ 2r$ $= \left(\frac{\theta}{360} \times 6{\cdot}2832 \times r\right) + 2r$	Area $= \frac{\theta}{360}\left(3{\cdot}1416 r^2\right)$
$=$ arc ABC $+ b$ *Note* : $\frac{b^2}{4} + (r-h)^2 = r^2$	Area $= \frac{\theta}{360}\left(3{\cdot}1416 r^2\right) - \frac{b}{2}\left(r-h\right)$
	Area $= a \times b \times 3{\cdot}1417$
$= a\left(1 + \frac{8}{3}\frac{b^2}{a^2}\right) + a$	Area $= \frac{2}{3} \cdot a \cdot b$

MENSURATION. OF

AREAS OF IRREGULAR FIGURES.





PLANE SURFACES

METHOD OF MIDDLE OR MEAN ORDINATES.

Draw two parallel lines AB, CD at opposite ends of the irregular figure, and select a base line BD at right angles to AB and CD. Divide the base into *n* equal parts (preferably 10), and from the middle of each part erect ordinates 1, 2, 3, 4, 5, etc., parallel to the lines AB, CD. The length of the ordinates cut off by the figures are y^1, y^2, y^3, etc.

$$\text{Now the Area} = \frac{l}{n} (y^1 + y^2 + y^3 + \ \ldots \ y^n)$$

SIMPSON'S METHOD.

Draw AB, CD as before at the extreme positions of the figure, and choose a base line BD at right angles to AB. Divide the base into *n* equal parts, and erect ordinates y_1, y_2, $y_3 \ldots y_n$. There will be an odd number of ordinates including the two end ones spaced apart

$$d = \frac{l}{n}$$

$$\text{The Area} = \frac{d}{3} \left\{ (y_1 + y_n) + 4\,(y_2 + y_4 + y_6 + \ \ldots \ y_{n-1}) + 2\,(y_3 + y_5 \ \ldots + y_{n-2}) \right\}$$

Expressed in words: The Area is obtained by adding together the first and last ordinate, four times the sum of the even ordinates, and twice the sum of the odd ordinates, and then multiplying this result by $\frac{d}{3}$.

TABLE 9.

CHORD AND ARC OF CIRCLE OF UNIT RADIUS FOR DEGREES OF ANGLE.

Degrees.	Chord.	Arc.	Degrees.	Chord.	Arc.
1	·0175	·01745	46	·7815	·80283
2	·0349	·03490	47	·7975	·82029
3	·0524	·05235	48	·8135	·83774
4	·0698	·06981	49	·8294	·85519
5	·0872	·08726	50	·8452	·87266
6	·1047	·10471	51	·8610	·89010
7	·1221	·12217	52	·8767	·90755
8	·1395	·13962	53	·8924	·92500
9	·1569	·15707	54	·9080	·94246
10	·1743	·17453	55	·9235	·95991
11	·1917	·19198	56	·9389	·97736
12	·2091	·20943	57	·9543	·99482
13	·2264	·22688	58	·9696	1·01227
14	·2437	·24434	59	·9848	1·02972
15	·2611	·26179	60	1·0000	1·04217
16	·2783	·27924	61	1·0151	1·06463
17	·2956	·29670	62	1·0301	1·08208
18	·3129	·31415	63	1·0450	1·09953
19	·3301	·33160	64	1·0598	1·11699
20	·3473	·34906	65	1·0746	1·13444
21	·3645	·36651	66	1·0893	1·15189
22	·3816	·38396	67	1·1039	1·16935
23	·3987	·40141	68	1·1184	1·18680
24	·4158	·41887	69	1·1328	1·20425
25	·4329	·43632	70	1·1472	1·22170
26	·4499	·45377	71	1·1614	1·23916
27	·4669	·47123	72	1·1756	1·25661
28	·4838	·48868	73	1·1896	1·27406
29	·5008	·50613	74	1·2036	1·29152
30	·5176	·52359	75	1·2175	1·30897
31	·5345	·54104	76	1·2313	1·32642
32	·5512	·55849	77	1·2450	1·34388
33	·5680	·57594	78	1·2586	1·36133
34	·5847	·59340	79	1·2722	1·37878
35	·6014	·61085	80	1·2856	1·39623
36	·6180	·62830	81	1·2989	1·41369
37	·6346	·64576	82	1·3121	1·43114
38	·6511	·66321	83	1·3252	1·44859
39	·6676	·68066	84	1·3383	1·46605
40	·6840	·69813	85	1·3512	1·48350
41	·7004	·71557	86	1·3640	1·50095
42	·7167	·73302	87	1·3767	1·51841
43	·7330	·75047	88	1·3893	1·53586
44	·7492	·76793	89	1·4018	1·55331
45	·7654	·78538	90	1·4142	1·57076

Table 10.

Area and Circumference of Circle
Diameter advancing from ·1 to 15 by ·1

Diam.	Circum.	Area	Diam.	Circum.	Area.
·1	·31416	·007854	·1	12·8805	13·2025
·2	·62832	·031416	·2	13·1947	13·8544
·3	·94248	·070686	·3	13·5088	14·5220
·4	1·2566	·12566	·4	13·8230	15·2053
·5	1·5708	·19635	·5	14·1372	15·9043
·6	1·8850	·28274	·6	14·4513	16·6190
·7	2·1991	·38485	·7	14·7655	17·3494
·8	2·5133	·50266	·8	15·0796	18·0956
·9	2·8274	·63617	·9	15·3938	18·8574
1·0	3·1416	·7854	5·0	15·7080	19·6350
·1	3·4558	·9503	·1	16·0221	20·4282
·2	3·7699	1·1310	·2	16·3363	21·2372
·3	4·0841	1·3273	·3	16·6504	22·0618
·4	4·3982	1·5394	·4	16·9646	22·9022
·5	4·7124	1·7671	·5	17·2788	23·7583
·6	5·0265	2·0106	·6	17·5929	24·6301
·7	5·3407	2·2698	·7	17·9071	25·5176
·8	5·6549	2·5447	·8	18·2212	26·4208
·9	5·9690	2·8353	·9	18·5354	27·3397
2·0	6·2832	3·1416	6·0	18·8496	28·2743
·1	6·5973	3·4636	·1	19·1637	29·2247
·2	6·9115	3·8013	·2	19·4779	30·1907
·3	7·2257	4·1548	·3	19·7920	31·1725
·4	7·5398	4·5239	·4	20·1062	32·1699
·5	7·8540	4·9087	·5	20·4204	33·1831
·6	8·1681	5·3093	·6	20·7345	34·2119
·7	8·4823	5·7256	·7	21·0487	35·2565
·8	8·7965	6·1575	·8	21·3628	36·3168
·9	9·1106	6·6052	·9	21·6770	37·3928
3·0	9·4248	7·0686	7·0	21·9911	38·4845
·1	9·7389	7·5477	·1	22·3053	39·5919
·2	10·0531	8·0425	·2	22·6195	40·7150
·3	10·3673	8·5530	·3	22·9336	41·8539
·4	10·6814	9·0792	·4	23·2478	43·0084
·5	10·9956	9·6211	·5	23·5619	44·1786
·6	11·3097	10·1788	·6	23·8761	45·3646
·7	11·6239	10·7521	·7	24·1903	46·5663
·8	11·9381	11·3411	·9	24·5044	47·7836
·9	12·2522	11·9459	·9	24·8186	49·0617
4·0	12·5664	12·5664	8·0	25·1327	50·2655

TABLE 10 (*Continued*).

Diam.	Circum.	Area.
·1	25·4469	51·5300
·2	25·7611	52·8102
·3	26·0752	54·1061
·4	26·3894	55·4177
·5	26·7035	56·7450
·6	27·0177	58·0880
·7	27·3319	59·4468
·8	27·6460	60·8212
·9	27·9602	62·2114
9·0	28·2743	63·6173
·1	28·5885	65·0388
·2	28·9027	66·4761
·3	29·2168	67·9291
·4	29·5310	69·3978
·5	29·8451	70·8822
·6	30·1593	72·3823
·7	30·4734	73·8981
·8	30·7876	75·4296
·9	31·1018	76·9769
10·0	31·4159	78·5398
·1	31·7301	80·1185
·2	32·0442	81·7128
·3	32·3584	83·3229
·4	32·6726	84·9487
·5	32·9867	86·5901
·6	33·3009	88·2473
·7	33·6150	89·9202
·8	33·9292	91·6088
·9	34·2434	93·3132
11·0	34·5575	95·0332
·1	34·8717	96·7689
·2	35·1858	98·5203
·3	35·5000	100·2875
·4	35·8142	102·0703
·5	36·1283	103·8689
·6	36·4425	105·6832
·7	36·7566	107·5132
·8	37·0708	109·3588
·9	37·3850	111·2202
12·0	37·6991	113·0973
·1	38·0133	114·9901
·2	38·3274	116·8987
·3	38·6416	118·8229
·4	38·9557	120·7628
·5	39·2699	122·7185
·6	39·5841	124·6898
·7	39·8982	126·6769
·8	40·2124	128·6796
·9	40·5265	130·6981
13·0	40·8407	132·7323
·1	41·1549	134·7822
·2	41·4690	136·8478
·3	41·7832	138·9291
·4	42·0973	141·0261
·5	42·4115	143·1388
·6	42·7257	145·2672
·7	43·0398	147·4114
·8	43·3540	149·5712
·9	43·6681	151·7468
14·0	43·9823	153·9380
·1	44·2965	156·1450
·2	44·6106	158·3677
·3	44·9248	160·6061
·4	45·2389	162·8602
·5	45·5531	165·1300
·6	45·8673	167·4155
·7	46·1814	169·7167
·8	46·4956	172·0336
·9	46·8097	174·3662
15·0	47·1239	176·7146

TABLE 11.

AREAS AND CIRCUMFERENCES OF CIRCLES IN INCHES.

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
$\frac{1}{64}$	0·00019	0·049087	1	0·78540	3·14159
$\frac{1}{32}$	0·00077	0·098175	$1\frac{1}{16}$	0·88664	3·33794
$\frac{3}{64}$	0·00173	0·147262	$1\frac{1}{8}$	0·99402	3·53429
$\frac{1}{16}$	0·00307	0·196350	$1\frac{3}{16}$	1·1075	3·73064
$\frac{5}{64}$	0·00480	0·245437	$1\frac{1}{4}$	1·2272	3·92699
$\frac{3}{32}$	0·00690	0·294524	$1\frac{5}{16}$	1·3530	4·12334
$\frac{7}{64}$	0·00940	0·343612	$1\frac{3}{8}$	1·4849	4·31969
$\frac{1}{8}$	0·01227	0·392699	$1\frac{7}{16}$	1·6230	4·51604
$\frac{9}{64}$	0·01550	0·441787	$1\frac{1}{2}$	1·7671	4·71239
$\frac{5}{32}$	0·01917	0·490874	$1\frac{9}{16}$	1·9175	4·90874
$\frac{11}{64}$	0·02320	0·539962	$1\frac{5}{8}$	2·0739	5·10609
$\frac{3}{16}$	0·02761	0·589049	$1\frac{11}{16}$	2·2365	5·30144
$\frac{13}{64}$	0·03400	0·638137	$1\frac{3}{4}$	2·4053	5·49779
$\frac{7}{32}$	0·03758	0·687223	$1\frac{13}{16}$	2·5802	5·69414
$\frac{15}{64}$	0·04310	0·736312	$1\frac{7}{8}$	2·7612	5·89049
$\frac{1}{4}$	0·04909	0·785398	$1\frac{15}{16}$	2·9483	6·08684
$\frac{9}{32}$	0·06213	0·883573			
$\frac{5}{16}$	0·07670	0·981748	2	3·1416	6·28319
$\frac{11}{32}$	0·09281	1·07992	$2\frac{1}{16}$	3·3410	6·47953
$\frac{3}{8}$	0·11045	1·17810	$2\frac{1}{8}$	3·5466	6·67588
$\frac{13}{32}$	0·12962	1·27627	$2\frac{3}{16}$	3·7583	6·87223
$\frac{7}{16}$	0·15033	1·37445	$2\frac{1}{4}$	3·9761	7·06858
$\frac{15}{32}$	0·17257	1·47262	$2\frac{5}{16}$	4·2000	7·26493
$\frac{1}{2}$	0·19635	1·57080	$2\frac{3}{8}$	4·4301	7·46128
$\frac{17}{32}$	0·22166	1·66897	$2\frac{7}{16}$	4·6664	7·65763
$\frac{9}{16}$	0·24850	1·76715	$2\frac{1}{2}$	4·9087	7·85398
$\frac{19}{32}$	0·27688	1·86532	$2\frac{9}{16}$	5·1572	8·05033
$\frac{5}{8}$	0·30680	1·96350	$2\frac{5}{8}$	5·4119	8·24668
$\frac{21}{32}$	0·33824	2·06167	$2\frac{11}{16}$	5·6727	8·44303
$\frac{11}{16}$	0·37122	2·15984	$2\frac{3}{4}$	5·9396	8·63938
$\frac{23}{32}$	0·40574	2·25802	$2\frac{13}{16}$	6·2126	8·83573
$\frac{3}{4}$	0·44179	2·35619	$2\frac{7}{8}$	6·4918	9·03208
$\frac{25}{32}$	0·47937	2·45437	$2\frac{15}{16}$	6·7771	9·22843
$\frac{13}{16}$	0·51849	2·55254			
$\frac{27}{32}$	0·55914	2·65072	3	7·0686	9·42478
$\frac{7}{8}$	0·60132	2·74889	$3\frac{1}{16}$	7·3662	9·62113
$\frac{29}{32}$	0·64504	2·84707	$3\frac{1}{8}$	7·6699	9·81748
$\frac{15}{16}$	0·69029	2·94524	$3\frac{3}{16}$	7·9798	10·0138
$\frac{31}{32}$	0·73706	3·04342	$3\frac{1}{4}$	8·2958	10·2102

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
$3\frac{5}{16}$	8·6179	10·4065	$5\frac{3}{4}$	25·967	18·0642
$3\frac{3}{8}$	8·9462	10·6029	$5\frac{13}{16}$	26·535	18·2610
$3\frac{7}{16}$	9·2806	10·7992	$5\frac{7}{8}$	27·109	18·4569
$3\frac{1}{2}$	9·6211	10·9956	$5\frac{15}{16}$	27·688	18·6530
$3\frac{9}{16}$	9·9678	11·1919			
$3\frac{5}{8}$	10·321	11·3883	6	28·274	18·8496
$3\frac{11}{16}$	10·680	11·5846	$6\frac{1}{16}$	28·866	19·0459
$3\frac{3}{4}$	11·045	11·7810	$6\frac{1}{8}$	29·464	19·2423
$3\frac{13}{16}$	11·416	11·9773	$6\frac{3}{16}$	30·069	19·4386
$3\frac{7}{8}$	11·793	12·1737	$6\frac{1}{4}$	30·679	19·6350
$3\frac{15}{16}$	12·177	12·3700	$6\frac{5}{16}$	31·296	19·8313
			$6\frac{3}{8}$	31·919	20·0277
4	12·566	12·5664	$6\frac{7}{16}$	32·548	20·2240
$4\frac{1}{16}$	12·962	12·7627	$6\frac{1}{2}$	33·183	20·4204
$4\frac{1}{8}$	13·364	12·9591	$6\frac{9}{16}$	33·824	20·6167
$4\frac{3}{16}$	13·772	13·1554	$6\frac{5}{8}$	34·471	20·8131
$4\frac{1}{4}$	14·186	13·3518	$6\frac{11}{16}$	35·125	21·0094
$4\frac{5}{16}$	14·607	13·5481	$6\frac{3}{4}$	35·784	21·2058
$4\frac{3}{8}$	15·033	13·7445	$6\frac{13}{16}$	36·450	21·4021
$4\frac{7}{16}$	15·466	13·9408	$6\frac{7}{8}$	37·122	21·5985
$4\frac{1}{2}$	15·904	14·1372	$6\frac{15}{16}$	37·800	21·7948
$4\frac{9}{16}$	16·349	14·3335			
$4\frac{5}{8}$	16·800	14·5299	7	38·484	21·9912
$4\frac{11}{16}$	17·257	14·7262	$7\frac{1}{8}$	39·871	22·3839
$4\frac{3}{4}$	17·721	14·9226	$7\frac{1}{4}$	41·282	22·7766
$4\frac{13}{16}$	18·190	15·1189	$7\frac{3}{8}$	42·718	23·1693
$4\frac{7}{8}$	18·665	15·3153	$7\frac{1}{2}$	44·178	23·5620
$4\frac{15}{16}$	19·147	15·5116	$7\frac{5}{8}$	45·663	23·9547
			$7\frac{3}{4}$	47·173	24·3474
5	19·635	15·7080	$7\frac{7}{8}$	48·707	24·7401
$5\frac{1}{16}$	20·129	15·9043			
$5\frac{1}{8}$	20·629	16·1007	8	50·265	25·1328
$5\frac{3}{16}$	21·135	16·2960	$8\frac{1}{8}$	51·848	25·5225
$5\frac{1}{4}$	21·648	16·4934	$8\frac{1}{4}$	53·456	25·9182
$5\frac{5}{16}$	22·166	16·6890	$8\frac{3}{8}$	55·088	26·3109
$5\frac{3}{8}$	22·690	16·8861	$8\frac{1}{2}$	56·745	26·7036
$5\frac{7}{16}$	23·221	17·0820	$8\frac{5}{8}$	58·426	27·0963
$5\frac{1}{2}$	23·758	17·2788	$8\frac{3}{4}$	60·132	27·4890
$5\frac{9}{16}$	24·301	17·4749	$8\frac{7}{8}$	61·682	27·8817
$5\frac{5}{8}$	24·850	17·6715			
$5\frac{11}{16}$	25·406	17·8670	9	63·617	28·2743

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
9⅛	65·396	28·6670	13¾	148·489	43·1969
9¼	67·200	29·0597	13⅞	151·201	43·5896
9⅜	69·029	29·4524			
9½	70·882	29·8451	14	153·938	43·9823
9⅝	72·759	30·2378	14⅛	156·699	44·3750
9¾	74·661	30·6305	14¼	159·485	44·7677
9⅞	76·588	31·0232	14⅜	162·295	45·1604
			14½	165·130	45·5531
10	78·539	31·4159	14⅝	167·989	45·9458
10⅛	80·515	31·8086	14¾	170·873	46·3385
10¼	82·515	32·2013	14⅞	173·782	46·7312
10⅜	84·540	32·5940			
10½	86·590	32·9867	15	176·715	47·1239
10⅝	88·664	33·3794	15⅛	179·672	47·5166
10¾	90·762	33·7721	15¼	182·654	47·9093
10⅞	92·885	34·1648	15⅜	185·661	48·3020
			15½	188·692	48·6947
11	95·033	34·5575	15⅝	191·748	49·0874
11⅛	97·205	34·9502	15¾	194·828	49·4801
11¼	99·402	35·3429	15⅞	197·933	49·8728
11⅜	101·623	35·7356			
11½	103·869	36·1283	16	201·062	50·2655
11⅝	106·139	36·5210	16⅛	204·216	50·6582
11¾	108·434	36·9137	16¼	207·394	51·0509
11⅞	110·753	37·3064	16⅜	210·597	51·4436
			16½	213·825	51·8363
12	113·097	37·6991	16⅝	217·077	52·2290
12⅛	115·466	38·0918	16¾	220·353	52·6217
12¼	117·859	38·4845	16⅞	223·654	53·0144
12⅜	120·276	38·8772			
12½	122·718	39·2699	17	226·980	53·4071
12⅝	125·184	39·6626	17⅛	230·330	53·7998
12¾	127·676	40·0553	17¼	233·705	54·1925
12⅞	130·192	40·4480	17⅜	237·104	54·5852
			17½	240·528	54·9779
13	132·732	40·8407	17⅝	243·977	55·3706
13⅛	135·297	41·2334	17¾	247·450	55·7633
13¼	137·886	41·6261	17⅞	250·947	56·1560
13⅜	140·500	42·0188			
13½	143·139	42·4115	18	254·469	56·5487
13⅝	145·802	42·8042	18⅛	258·016	56·9414

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
18¼	261·587	57·3341	22⅞	410·972	71·8639
18⅜	265·182	57·7268			
18½	268·803	58·1195	23	415·476	72·2566
18⅝	272·447	58·5122	23⅛	420·004	72·6493
18¾	276·117	58·9049	23¼	424·557	73·0420
18⅞	279·811	59·2976	23⅜	429·135	73·4347
			23½	433·731	73·8274
19	283·529	59·6903	23⅝	438·363	74·2201
19⅛	287·272	60·0830	23¾	443·014	74·6128
19¼	291·039	60·4757	23⅞	447·699	75·0054
19⅜	294·831	60·8684			
19½	298·648	61·2611	24	452·390	75·3928
19⅝	302·489	61·6538	24⅛	457·115	75·7909
19¾	306·355	62·0465	24¼	461·864	76·1836
19⅞	310·245	62·4392	24⅜	466·638	76·5763
			24½	471·436	76·9690
20	314·160	62·8319	24⅝	476·258	77·3617
20⅛	318·099	63·2245	24¾	481·106	77·7544
20¼	322·063	63·6173	24⅞	485·978	78·1471
20⅜	326·051	64·0100			
20½	330·064	64·4026	25	490·875	78·5398
20⅝	334·101	64·7953	25⅛	495·796	78·9325
20¾	338·163	65·1880	25¼	500·741	79·3252
20⅞	342·250	65·5807	25⅜	505·711	79·7179
			25½	510·706	80·1106
21	346·361	65·9734	25⅝	515·725	80·5033
21⅛	350·497	66·3661	25¾	520·769	80·8960
21¼	354·657	66·7588	25⅞	525·837	81·2887
21⅜	358·841	67·1515			
21½	363·051	67·5442	26	530·930	81·6814
21⅝	367·284	67·9369	26⅛	536·047	82·0741
21¾	371·543	68·3296	26¼	541·189	82·4668
21⅞	375·826	68·7223	26⅜	546·356	82·8595
			26½	551·547	83·2522
22	380·133	69·1150	26⅝	556·762	83·6449
22⅛	384·465	69·5077	26¾	562·002	84·0376
22¼	388·822	69·9004	26⅞	567·267	84·4303
22⅜	393·203	70·2931			
22½	397·608	70·6858	27	572·556	84·8230
22⅝	402·038	71·0785	27⅛	577·870	85·2157
22¾	406·493	71·4712	27¼	583·208	85·6084

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
27⅜	588·571	86·0011	32	804·249	100·531
27½	593·958	86·3938	32⅛	810·545	100·923
27⅝	599·370	86·7865	32¼	816·865	101·316
27¾	604·807	87·1792	32⅜	823·209	101·709
27⅞	610·268	87·5719	32½	829·578	102·101
			32⅝	835·972	102·494
28	615·753	87·9646	32¾	842·390	102·887
28⅛	621·263	88·3573	32⅞	848·833	103·279
28¼	626·798	88·7500			
28⅜	632·357	89·1427	33	855·29	103·672
28½	637·941	89·5354	33⅛	861·79	104·065
28⅝	643·594	89·9281	33¼	868·30	104·458
28¾	649·182	90·3208	33⅜	874·84	104·850
28⅞	654·839	90·7135	33½	881·41	105·243
			33⅝	888·00	105·636
29	660·521	91·1062	33¾	894·61	106·028
29⅛	666·227	91·4989	33⅞	901·25	106·421
29¼	671·958	91·8916			
29⅜	677·714	92·2843	34	907·92	106·814
29½	683·494	92·6770	34⅛	914·60	107·206
29⅝	689·298	93·0697	34¼	921·32	107·599
29¾	695·128	93·4624	34⅜	928·05	107·992
29⅞	700·981	93·8551	34½	934·82	108·384
			34⅝	941·60	108·788
30	706·860	94·2478	34¾	948·41	109·170
30⅛	712·762	94·6405	34⅞	955·25	109·563
30¼	718·690	95·0332			
30⅜	724·641	95·4259	35	962·11	109·955
30½	730·618	95·8186	35⅛	968·99	110·348
30⅝	736·619	96·2113	35¼	975·90	110·741
30¾	742·644	96·6040	35⅜	982·83	111·133
30⅞	748·694	96·9967	35½	989·79	111·526
			35⅝	996·78	111·919
31	754·769	97·3894	35¾	1003·78	112·311
31⅛	760·868	97·7821	35⅞	1010·81	112·704
31¼	766·992	98·1748			
31⅜	773·140	98·5675	36	1017·87	113·097
31½	779·313	98·9602	36⅛	1024·95	113·490
31⅝	785·510	99·3529	36¼	1032·06	113·882
31¾	791·732	99·7456	36⅜	1039·19	114·275
31⅞	797·978	100·138	36½	1046·34	114·668

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
36⅝	1053·52	115·060	41⅛	1328·31	129·198
36¾	1060·72	115·453	41¼	1336·40	129·590
36⅞	1067·95	115·846	41⅜	1344·51	129·993
			41½	1352·65	130·376
37	1075·21	116·238	41⅝	1360·81	130·768
37⅛	1082·48	116·631	41¾	1368·99	131·161
37¼	1089·78	117·024	41⅞	1377·20	131·554
37⅜	1097·11	117·417			
37½	1104·46	117·809	42	1385·44	131·946
37⅝	1111·84	118·202	42⅛	1393·70	132·339
37¾	1119·24	118·595	42¼	1401·98	132·732
37⅞	1126·66	118·987	42⅜	1410·29	133·125
			42½	1418·62	133·517
38	1134·11	119·380	42⅝	1426·98	133·910
38⅛	1141·58	119·773	42¾	1435·36	134·303
38¼	1149·08	120·165	42⅞	1443·77	134·695
38⅜	1156·60	120·558			
38½	1164·15	120·951	43	1452·20	135·088
38⅝	1171·72	121·344	43⅛	1460·65	135·481
38¾	1179·32	121·736	43¼	1469·13	135·873
38⅞	1186·94	122·129	43⅜	1477·64	136·266
			43½	1486·16	136·659
39	1194·59	122·522	43⅝	1494·72	137·052
39⅛	1202·26	122·914	43¾	1503·30	137·444
39¼	1209·95	123·307	43⅞	1511·90	137·837
39⅜	1217·67	123·700			
39½	1225·41	124·092	44	1520·53	138·230
39⅝	1233·18	124·485	44⅛	1529·18	138·622
39¾	1240·97	124·878	44¼	1537·85	139·015
39⅞	1248·79	125·271	44⅜	1546·55	139·408
			44½	1555·28	139·800
40	1256·63	125·663	44⅝	1564·03	140·193
40⅛	1264·50	126·056	44¾	1572·80	140·586
40¼	1272·39	126·449	44⅞	1581·60	140·979
40⅜	1280·30	126·841			
40½	1288·24	127·234	45	1590·43	141·371
40⅝	1296·21	127·627	45⅛	1599·27	141·764
40¾	1304·20	128·019	45¼	1608·15	142·157
40⅞	1312·21	128·412	45⅜	1617·04	142·549
			45½	1625·97	142·942
41	1320·25	128·805	45⅝	1634·91	143·335

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
45¾	1643·88	143·729	50¼	1983·2	157·865
45⅞	1652·88	144·120	50⅜	1993·1	158·258
			50½	2003·0	158·650
46	1661·90	144·513	50⅝	2012·9	159·043
46⅛	1670·94	144·906	50¾	2022·8	159·436
46¼	1680·01	145·298	50⅞	2032·8	159·829
46⅜	1689·10	145·691			
46½	1698·22	146·084	51	2042·8	160·221
46⅝	1707·36	146·476	51⅛	2052·8	160·614
46¾	1716·53	146·869	51¼	2062·9	161·007
46⅞	1725·72	147·262	51⅜	2073·0	161·399
			51½	2083·1	161·792
47	1734·94	147·654	51⅝	2093·2	162·186
47⅛	1744·18	148·047	51¾	2103·3	162·577
47¼	1753·45	148·440	51⅞	2113·5	162·970
47⅜	1762·74	148·832			
47½	1772·05	149·225	52	2123·7	163·363
47⅝	1781·39	149·618	52⅛	2133·9	163·756
47¾	1790·75	150·011	52¼	2144·2	164·148
47⅞	1800·14	150·403	52⅜	2154·5	164·541
			52½	2164·8	164·934
48	1809·55	150·796	52⅝	2175·1	165·326
48⅛	1818·99	151·189	52¾	2185·4	165·719
48¼	1828·45	151·581	52⅞	2195·8	166·112
48⅜	1837·94	151·974			
48½	1847·45	152·367	53	2206·2	166·504
48⅝	1856·98	152·759	53⅛	2216·6	166·897
48¾	1866·54	153·152	53¼	2227·0	167·290
48⅞	1876·13	153·545	53⅜	2237·5	167·683
			53½	2248·0	168·075
49	1885·74	153·938	53⅝	2258·5	168·468
49⅛	1895·37	154·330	53¾	2269·1	168·861
49¼	1905·03	154·723	53⅞	2279·6	169·253
49⅜	1914·71	155·116			
49½	1924·42	155·508	54	2290·2	169·646
49⅝	1934·15	155·904	54⅛	2300·8	170·039
49¾	1943·90	156·294	54¼	2311·5	170·431
49⅞	1953·69	156·686	54⅜	2322·1	170·824
			54½	2332·8	171·217
50	1963·49	157·079	54⅝	2343·5	171·609
50⅛	1973·3	157·472	54¾	2354·3	172·002

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
54⅞	2365·0	172·395	59⅜	2768·8	186·532
			59½	2780·5	186·925
55	2375·8	172·788	59⅝	2792·2	187·317
55⅛	2386·6	173·180	59¾	2803·9	187·710
55¼	2397·5	173·573	59⅞	2815·7	188·103
55⅜	2408·3	173·966			
55½	2419·2	174·358	60	2827·4	188·496
55⅝	2430·1	174·751	60⅛	2839·2	188·888
55¾	2441·1	175·144	60¼	2851·0	189·281
55⅞	2452·0	175·536	60⅜	2862·9	189·674
			60½	2874·8	190·066
56	2463·0	175·929	60⅝	2886·6	190·459
56⅛	2474·0	176·322	60¾	2898·6	190·852
56¼	2485·0	176·715	60⅞	2910·5	191·244
56⅜	2496·1	177·107			
56½	2507·2	177·500	61	2922·5	191·637
56⅝	2518·3	177·893	61⅛	2934·5	192·030
56¾	2529·4	178·285	61¼	2946·5	192·423
56⅞	2540·6	178·678	61⅜	2958·5	192·815
			61½	2970·6	193·208
57	2551·8	179·071	61⅝	2982·7	193·601
57⅛	2563·0	179·463	61¾	2994·8	193·993
57¼	2574·2	179·856	61⅞	3006·9	194·386
57⅜	2585·4	180·249			
57½	2596·7	180·642	62	3019·1	194·779
57⅝	2608·0	181·034	62⅛	3031·3	195·171
57¾	2619·4	181·427	62¼	3043·5	195·564
57⅞	2630·7	181·820	62⅜	3055·7	195·957
			62½	3068·0	196·350
58	2642·1	182·212	62⅝	3080·3	196·742
58⅛	2653·5	182·605	62¾	3092·6	197·135
58¼	2664·9	182·998	62⅞	3104·9	197·528
58⅜	2676·4	183·390			
58½	2687·8	183·783	63	3117·2	197·920
58⅝	2699·3	184·176	63⅛	3129·6	198·313
58¾	2710·9	184·569	63¼	3142·0	198·706
58⅞	2722·4	184·961	63⅜	3154·5	199·098
			63½	3166·9	199·491
59	2734·0	185·354	63⅝	3179·4	199·884
59⅛	2745·6	185·747	63¾	3191·9	200·277
59¼	2757·2	186·139	63⅞	3204·4	200·669

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
64	3217·0	201·062	68⅝	3698·7	215·592
64⅛	3229·6	201·455	68¾	3712·2	215·984
64¼	3242·2	201·847	68⅞	3725·7	216·377
64⅜	3254·8	202·240			
64½	3267·5	202·633	69	3739·3	216·770
64⅝	3280·1	203·025	69⅛	3752·8	217·163
64¾	3292·8	203·418	69¼	3766·4	217·555
64⅞	3305·6	203·811	69⅜	3780·0	217·948
			69½	3793·7	218·341
65	3318·3	204·204	69⅝	3807·3	218·733
65⅛	3331·1	204·596	69¾	3821·0	219·126
65¼	3343·9	204·989	69⅞	3834·7	219·519
65⅜	3356·7	205·382			
65½	3369·6	205·744	70	3848·5	219·911
65⅝	3382·4	206·167	70⅛	3862·2	220·304
65¾	3395·3	206·560	70¼	3876·0	220·697
65⅞	3408·2	206·952	70⅜	3889·8	221·090
			70½	3903·6	221·482
66	3421·2	207·345	70⅝	3917·5	221·875
66⅛	3434·3	207·738	70¾	3931·4	222·268
66¼	3447·2	208·131	70⅞	3945·3	222·660
66⅜	3460·2	208·523			
66½	3473·2	208·916	71	3959·2	223·053
66⅝	3486·3	209·309	71⅛	3973·1	223·446
66¾	3499·4	209·701	71¼	3987·1	223·838
66⅞	3512·5	210·094	71⅜	4001·1	224·231
			71½	4015·2	224·624
67	3525·7	210·487	71⅝	4029·2	225·017
67⅛	3538·8	210·879	71¾	4043·3	225·409
67¼	3552·0	211·272	71⅞	4057·4	225·802
67⅜	3565·2	211·665			
67½	3578·5	212·058	72	4071·5	226·195
67⅝	3591·7	212·450	72⅛	4085·7	226·587
67¾	3605·0	212·843	72¼	4099·8	226·980
67⅞	3618·3	213·236	72⅜	4114·0	227·373
			72½	4128·2	227·765
68	3631·7	213·628	72⅝	4142·5	228·158
68⅛	3645·0	214·021	72¾	4156·8	228·551
68¼	3658·4	214·414	72⅞	4171·8	228·944
68⅜	3671·8	214·806			
68½	3685·3	215·199	73	4185·4	229·336

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
73⅛	4199·7	229·729	77¾	4747·8	244·259
73¼	4214·1	230·122	77⅞	4763·1	244·652
73⅜	4228·5	230·514			
73½	4242·9	230·907	78	4778·4	245·044
73⅝	4257·4	231·300	78⅛	4793·7	245·437
73¾	4271·8	231·692	78¼	4809·0	245·830
73⅞	4286·3	232·085	78⅜	4824·4	246·222
			78½	4839·8	246·615
74	4300·8	232·478	78⅝	4855·2	247·008
74⅛	4315·4	232·871	78¾	4870·7	247·400
74¼	4329·9	233·263	78⅞	4886·2	247·793
74⅜	4344·5	233·656			
74½	4359·2	234·049	79	4901·7	248·186
74⅝	4373·8	234·441	79⅛	4917·2	248·579
74¾	4388·5	234·834	79¼	4932·7	248·971
74⅞	4403·1	235·227	79⅜	4948·3	249·364
			79½	4963·9	249·757
75	4417·9	235·619	79⅝	4979·5	250·149
75⅛	4432·6	236·012	79¾	4995·2	250·542
75¼	4447·4	236·405	79⅞	5010·9	250·935
75⅜	4462·2	236·798			
75½	4477·0	237·190	80	5026·5	251·327
75⅝	4491·8	237·583	80⅛	5042·3	251·720
75¾	4506·7	237·976	80¼	5058·0	252·113
75⅞	4521·5	238·368	80⅜	5073·8	252·506
			80½	5089·6	252·898
76	4536·5	238·761	80⅝	5105·4	253·291
76⅛	4551·4	239·154	80¾	5121·2	253·684
76¼	4566·4	239·546	80⅞	5137·1	254·076
76⅜	4581·3	239·939			
76½	4596·3	240·332	81	5153·0	254·469
76⅝	4611·4	240·725	81⅛	5168·9	254·862
76¾	4626·4	241·117	81¼	5184·9	255·254
76⅞	4641·5	241·510	81⅜	5200·8	255·647
			81½	5216·8	256·040
77	4656·6	241·903	81⅝	5232·8	256·433
77⅛	4671·8	242·295	81¾	5248·9	256·825
77¼	4686·9	242·688	81⅞	5264·9	257·218
77⅜	4702·1	243·081			
77½	4717·3	243·473	82	5281·0	257·611
77⅝	4732·5	243·866	82⅛	5297·1	258·003

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
82¼	5313·3	258·396	86⅞	5927·6	272·926
82⅜	5329·4	258·789			
82½	5345·6	259·181	87	5944·7	273·319
82⅝	5361·8	259·574	87⅛	5961·8	273·711
82¾	5378·1	259·967	87¼	5978·9	274·104
82⅞	5394·3	260·359	87⅜	5996·0	274·497
			87½	6013·2	274·889
83	5410·6	260·752	87⅝	6030·4	275·282
83⅛	5426·9	261·145	87¾	6047·6	275·675
83¼	5443·3	261·538	87⅞	6064·9	276·067
83⅜	5459·6	261·930			
83½	5476·0	262·323	88	6082·1	276·460
83⅝	5492·4	262·716	88⅛	6099·4	276·853
83¾	5508·8	263·108	88¼	6116·7	277·246
83⅞	5525·3	263·501	88⅜	6134·1	277·638
			88½	6151·4	278·031
84	5541·8	263·894	88⅝	6168·8	278·424
84⅛	5558·3	264·286	88¾	6186·2	278·816
84¼	5574·8	264·679	88⅞	6203·7	279·209
84⅜	5591·4	265·072			
84½	5607·9	265·465	89	6221·1	279·602
84⅝	5624·5	265·857	89⅛	6238·6	279·994
84¾	5641·2	266·250	89¼	6256·1	280·387
84⅞	5657·8	266·643	89⅜	6273·7	280·780
			89½	6291·2	281·173
85	5674·5	267·035	89⅝	6308·8	281·565
85⅛	5691·2	267·428	89¾	6326·4	281·958
85¼	5707·9	267·821	89⅞	6344·1	282·351
85⅜	5724·7	268·213			
85½	5741·5	268·606	90	6361·7	282·743
85⅝	5758·3	268·999	90⅛	6379·4	283·136
85¾	5775·1	269·392	90¼	6397·1	283·529
85⅞	5791·9	269·784	90⅜	6414·9	283·921
			90½	6432·6	284·314
86	5808·8	270·177	90⅝	6450·4	284·707
86⅛	5825·7	270·570	90¾	6468·2	285·100
86¼	5842·6	270·962	90⅞	6486·0	285·492
86⅜	5859·6	271·355			
86½	5876·5	271·748	91	6503·9	285·885
86⅝	5893·5	272·140	91⅛	6521·8	286·278
86¾	5910·6	272·533	91¼	6539·7	286·670

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
91⅜	6557·6	287·063	96	7238·2	301·593
91½	6575·5	287·456	96⅛	7257·1	301·986
91⅝	6593·5	287·848	96¼	7276·0	302·378
91¾	6611·5	288·241	96⅜	7294·9	302·771
91⅞	6629·6	288·634	96½	7313·8	303·164
			96⅝	7332·8	303·556
92	6647·6	289·027	96¾	7351·8	303·949
92⅛	6665·7	289·419	96⅞	7370·8	304·342
92¼	6683·8	289·812			
92⅜	6701·9	290·206	97	7389·8	304·734
92½	6720·1	290·597	97⅛	7408·9	305·127
92⅝	6738·2	290·990	97¼	7428·0	305·520
92¾	6756·4	291·383	97⅜	7447·1	305·913
92⅞	6774·7	291·775	97½	7466·8	306·305
			97⅝	7485·3	306·698
93	6792·9	292·168	97¾	7504·5	307·091
93⅛	6811·2	292·561	97⅞	7523·7	307·483
93¼	6829·5	292·954			
93⅜	6847·8	293·346	98	7543·0	307·876
93½	6866·1	293·739	98⅛	7562·2	308·269
93⅝	6884·5	294·132	98¼	7581·5	308·661
93¾	6902·9	294·524	98⅜	7600·8	309·054
93⅞	6921·3	294·917	98½	7620·1	309·447
			98⅝	7639·5	309·840
94	6939·8	295·310	98¾	7658·9	310·232
94⅛	6958·2	295·702	98⅞	7678·3	310·625
94¼	6976·7	296·095			
94⅜	6995·3	296·488	99	7697·7	311·018
94½	7013·8	296·881	99⅛	7717·1	311·410
94⅝	7032·4	297·273	99¼	7736·6	311·803
94¾	7051·0	297·666	99⅜	7756·1	312·196
94⅞	7069·6	298·059	99½	7775·6	312·588
			99⅝	7795·2	312·981
95	7088·2	298·451	99¾	7814·8	313·374
95⅛	7106·9	298·844	99⅞	7834·4	313·767
95¼	7125·6	299·237			
95⅜	7144·3	299·629	100	7853·9	314·159
95½	7163·0	300·022	100½	7938·7	315·730
95⅝	7181·8	300·415	101	8011·8	317·301
95¾	7200·6	300·807	101½	8091·3	318·872
95⅞	7219·4	301·200	102	8171·2	320·442

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
102½	8251·6	322·014	124	12076·2	389·56
103	8332·2	323·584	125	12271·8	392·70
103½	8413·4	325·154	126	12468·9	395·84
104	8494·8	326·726	127	12667·6	398·98
104½	8576·7	328·296	128	12867·9	402·12
105	8659·0	329·867	129	13069·8	405·26
105½	8741·6	331·438			
106	8824·7	333·009	130	13273·2	408·41
106½	8908·2	334·580	131	13478·2	411·55
107	8992·0	336·150	132	13684·7	414·69
107½	9076·2	337·722	133	13892·9	417·83
108	9160·8	339·292	134	14102·6	420·97
108½	9245·9	340·862	135	14313·8	424·11
109	9331·3	342·434	136	14526·7	427·26
109½	9417·1	344·004	137	14741·1	430·40
			138	14957·1	433·54
110	9503·3	345·575	139	15174·6	436·6
110½	9589·9	347·146			
111	9676·8	348·717	140	15393·8	439·8
111½	9764·2	350·288	141	15614·5	442·9
112	9852·0	351·858	142	15836·7	446·1
112½	9940·2	353·430	143	16060·6	449·2
113	10028·7	355·000	144	16286·0	452·3
113½	10117·6	356·570	145	16513·0	455·5
114	10207·0	358·142	146	16741·5	458·6
114½	10296·7	359·712	147	16971·6	461·8
115	10386·8	361·283	148	17203·3	464·9
115½	10477·4	362·854	149	17436·6	468·1
116	10568·3	364·425			
116½	10659·6	365·996	150	17671·4	471·2
117	10751·3	367·566	151	17907·8	474·3
117½	10843·4	369·138	152	18145·8	477·5
118	10935·8	370·708	153	18385·3	480·6
118½	11028·7	372·278	154	18626·5	483·8
119	11122·0	373·849	155	18869·1	486·9
119½	11215·6	375·420	156	19113·4	490·0
			157	19359·2	493·2
120	11309·7	376·991	158	19606·6	496·37
121	11499·0	380·13	159	19855·6	499·51
122	11689·8	383·27			
123	11882·2	386·42	160	20106·1	502·65

TABLE 11 (*Continued*).

Diameter in inches.	Area in square inches.	Circumference in inches.	Diameter in inches.	Area in square inches.	Circumference in inches.
161	20358·3	505·80	181	25730·4	568·63
162	20611·9	508·94	182	26015·5	571·77
163	20867·2	512·08	183	26302·2	574·91
164	21124·0	515·22	184	26590·4	578·05
165	21382·4	518·36	185	26880·2	581·19
166	21642·4	521·50	186	27171·6	584·34
167	21903·9	524·65	187	27464·5	587·48
168	22167·0	527·79	188	27759·1	590·62
169	22431·7	530·93	189	28055·2	593·76
170	22698·0	534·07	190	28352·8	596·90
171	22965·8	537·21	191	28652·1	600·04
172	23235·2	540·35	192	28952·9	603·19
173	23506·1	543·50	193	29255·3	606·33
174	23778·7	546·64	194	29559·2	609·47
175	24052·8	549·78	195	29864·7	612·61
176	24328·4	552·92	196	30171·8	615·75
177	24605·7	556·06	197	30480·5	618·89
178	24884·5	559·20	198	30790·7	622·03
179	25164·9	562·34	199	31102·5	625·18
180	25446·9	565·49	200	31415·9	628·32

Table 12.

Areas and Circumferences of Circles.

Diameter in feet & ins.	Circumference in feet & ins.		Areas in feet.	Diameter. in feet & ins.	Circumference in feet & ins.		Areas in feet.
1	0	3 1/8	0·0055	5 1/2	1	5 1/4	0·1649
1 1/8	0	3 1/2	0·0069	5 5/8	1	5 11/16	0·1725
1 1/4	0	3 7/8	0·0085	5 3/4	1	6 1/16	0·1802
1 3/8	0	4 1/4	0·0305	5 7/8	1	6 7/16	0·1881
1 1/2	0	4 5/8	0·0126				
1 5/8	0	5 1/8	0·0144	6	1	6 7/8	0·1963
1 3/4	0	5 1/2	0·0167	6 1/8	1	7 1/4	0·2044
1 7/8	0	5 7/8	0·0191	6 1/4	1	7 5/8	0·2130
				6 3/8	1	8	0·2216
2	0	6 1/4	0·0218	6 1/2	1	8 7/16	0·2304
2 1/8	0	6 5/8	0·0246	6 5/8	1	8 13/16	0·2393
2 1/4	0	7	0·0276	6 3/4	1	9 3/16	0·2484
2 3/8	0	7 3/8	0·0307	6 7/8	1	9 9/16	0·2577
2 1/2	0	7 3/4	0·0340				
2 5/8	0	8 1/4	0·0375	7	1	10	0·2669
2 3/4	0	8 5/8	0·0412	7 1/8	1	10 3/8	0·2768
2 7/8	0	9	0·0450	7 1/4	1	10 3/4	0·2865
				7 3/8	1	11 3/16	0·2966
3	0	9 3/8	0·0491	7 1/2	1	11 9/16	0·3067
3 1/8	0	9 3/4	0·0532	7 5/8	1	11 15/16	0·3170
3 1/4	0	10 1/4	0·0576	7 3/4	2	0 3/8	0·3275
3 3/8	0	10 5/8	0·0621	7 7/8	2	0 3/4	0·3381
3 1/2	0	11	0·0668				
3 5/8	0	11 3/8	0·0716	8	2	1 1/8	0·3490
3 3/4	0	11 3/4	0·0767	8 1/8	2	1 1/2	0·3600
3 7/8	1	0 1/8	0·0812	8 1/4	2	1 15/16	0·3711
				8 3/8	2	2 5/16	0·3825
4	1	0 9/16	0·0872	8 1/2	2	2 11/16	0·3937
4 1/8	1	0 15/16	0·0928	8 5/8	2	3 1/8	0·4056
4 1/4	1	1 3/8	0·0985	8 3/4	2	3 1/2	0·4189
4 3/8	1	1 3/4	0·1044	8 7/8	2	3 7/8	0·4298
4 1/2	1	2 1/8	0·1104				
4 5/8	1	2 9/16	0·1166	9	2	4 1/4	0·4417
4 3/4	1	2 15/16	0·1230	9 1/8	2	4 11/16	0·4540
4 7/8	1	3 5/16	0·1296	9 1/4	2	5 1/16	0·4666
				9 3/8	2	5 7/16	0·4793
5	1	3 11/16	0·1365	9 1/2	2	5 13/16	0·4922
5 1/8	1	4 1/8	0·1432	9 5/8	2	6 1/4	0·5052
5 1/4	1	4 1/2	0·1502	9 3/4	2	6 5/8	0·5177
5 3/8	1	4 7/8	0·1575	9 7/8	2	7	0·5318

TABLE 12 (*Continued*).

Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
	10	2	$7\frac{7}{16}$	0·5454	1	$4\frac{1}{2}$	4	$3\frac{13}{16}$	1·484
	$10\frac{1}{8}$	2	$7\frac{13}{16}$	0·5590	1	$4\frac{3}{4}$	4	$4\frac{5}{8}$	1·529
	$10\frac{1}{4}$	2	$8\frac{1}{4}$	0·5730					
	$10\frac{3}{8}$	2	$8\frac{9}{16}$	0·5870	1	5	4	$5\frac{7}{16}$	1·575
	$10\frac{1}{2}$	2	9	0·6013	1	$5\frac{1}{4}$	4	$6\frac{5}{16}$	1·622
	$10\frac{5}{8}$	2	$9\frac{3}{8}$	0·6156	1	$5\frac{1}{2}$	4	7	1·670
	$10\frac{3}{4}$	2	$9\frac{3}{4}$	0·6302	1	$5\frac{3}{4}$	4	$7\frac{3}{4}$	1·718
	$10\frac{7}{8}$	2	$10\frac{3}{16}$	0·6450					
					1	6	4	$8\frac{9}{16}$	1·766
	11	2	$10\frac{9}{16}$	0·6599	1	$6\frac{1}{4}$	4	$9\frac{3}{8}$	1·816
	$11\frac{1}{8}$	2	$10\frac{15}{16}$	0·6750	1	$6\frac{1}{2}$	4	$10\frac{1}{8}$	1·867
	$11\frac{1}{4}$	2	$11\frac{5}{16}$	0·6902	1	$6\frac{3}{4}$	4	11	1·917
	$11\frac{3}{8}$	2	$11\frac{3}{4}$	0·7055					
	$11\frac{1}{2}$	3	$0\frac{1}{8}$	0·7208	1	7	4	$11\frac{11}{16}$	1·968
	$11\frac{5}{8}$	3	$0\frac{1}{2}$	0·7368	1	$7\frac{1}{4}$	5	$0\frac{1}{2}$	2·020
	$11\frac{3}{4}$	3	$0\frac{15}{16}$	0·7528	1	$7\frac{1}{2}$	5	$1\frac{1}{4}$	2·073
	$11\frac{7}{8}$	3	$1\frac{3}{16}$	0·7687	1	$7\frac{3}{4}$	5	2	2·127
1	0	3	$1\frac{11}{16}$	0·7847	1	8	5	$2\frac{13}{16}$	2·181
1	$0\frac{1}{4}$	3	$2\frac{1}{2}$	0·8180	1	$8\frac{1}{4}$	5	$3\frac{5}{8}$	2·236
1	$0\frac{1}{2}$	3	$3\frac{1}{4}$	0·8520	1	$8\frac{1}{2}$	5	$4\frac{1}{2}$	2·291
1	$0\frac{3}{4}$	3	$4\frac{1}{16}$	0·8859	1	$8\frac{3}{4}$	5	$5\frac{1}{8}$	2·347
1	1	3	$4\frac{7}{8}$	0·9215	1	9	5	6	2·404
1	$1\frac{1}{4}$	3	$5\frac{5}{8}$	0·9568	1	$9\frac{1}{4}$	5	$6\frac{3}{4}$	2·462
1	$1\frac{1}{2}$	3	$6\frac{7}{16}$	0·9937	1	$9\frac{1}{2}$	5	$7\frac{1}{2}$	2·520
1	$1\frac{3}{4}$	3	$7\frac{1}{8}$	1·030	1	$9\frac{3}{4}$	5	$8\frac{5}{16}$	2·580
1	2	3	8	1·068	1	10	5	$9\frac{1}{8}$	2·640
1	$2\frac{1}{4}$	3	$8\frac{3}{4}$	1·106	1	$10\frac{1}{4}$	5	10	2·700
1	$2\frac{1}{2}$	3	$9\frac{9}{16}$	1·146	1	$10\frac{1}{2}$	5	$10\frac{11}{16}$	2·761
1	$2\frac{3}{4}$	3	$10\frac{5}{16}$	1·186	1	$10\frac{3}{4}$	5	$11\frac{1}{2}$	2·822
1	3	3	$11\frac{1}{8}$	1·227	1	11	6	$0\frac{1}{4}$	2·884
1	$3\frac{1}{4}$	3	$11\frac{15}{16}$	1·268	1	$11\frac{1}{4}$	6	1	2·947
1	$3\frac{1}{2}$	4	$0\frac{11}{16}$	1·309	1	$11\frac{1}{2}$	6	$1\frac{13}{16}$	3·011
1	$3\frac{3}{4}$	4	$1\frac{1}{2}$	1·352	1	$11\frac{3}{4}$	6	$2\frac{5}{8}$	3·076
1	4	4	$2\frac{1}{4}$	1·396	2	0	6	$3\frac{3}{8}$	3·141
1	$4\frac{1}{4}$	4	3	1·438	2	$0\frac{1}{4}$	6	$4\frac{1}{8}$	3·207

TABLE 12 (*Continued*).

Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
2	0½	6	4⅞	3·273	2	8½	8	6⅛	5·760
2	0¾	6	5¾	3·341	2	8¾	8	6⅞	5·849
2	1	6	6½	3·408	2	9	8	7⅝	5·939
2	1¼	6	7¼	3·477	2	9¼	8	8½	6·029
2	1½	6	8⅛	3·546	2	9½	8	9¼	6·120
2	1¾	6	8⅞	3·610	2	9¾	8	10	6·212
2	2	6	9⅝	3·687	2	10	8	10¾	6·305
2	2¼	6	10½	3·758	2	10¼	8	11½	6·398
2	2½	6	11¼	3·830	2	10½	9	0⅜	6·491
2	2¾	7	0	3·904	2	10¾	9	1⅛	6·586
2	3	7	0¾	3·976	2	11	9	1⅞	6·681
2	3¼	7	1⅝	4·050	2	11¼	9	2¾	6·777
2	3½	7	2⅜	4·124	2	11½	9	3½	6·873
2	3¾	7	3⅛	4·200	2	11¾	9	4¼	6·970
2	4	7	3⅞	4·276	3	0	9	5	7·068
2	4¼	7	4⅝	4·352	3	0¼	9	5⅞	7·167
2	4½	7	5½	4·430	3	0½	9	6⅝	7·266
2	4¾	7	6¼	4·508	3	0¾	9	7½	7·366
2	5	7	7	4·586	3	1	9	8¼	7·466
2	5¼	7	7⅞	4·666	3	1¼	9	9	7·567
2	5½	7	8⅝	4·746	3	1½	9	9⅞	7·669
2	5¾	7	9½	4·827	3	1¾	9	10½	7·779
2	6	7	10¼	4·908	3	2	9	11⅜	7·868
2	6¼	7	11	4·990	3	2¼	10	0⅛	7·979
2	6½	7	11¾	5·073	3	2½	10	0⅞	8·084
2	6¾	8	0⅝	5·157	3	2¾	10	1¾	8·189
2	7	8	1⅜	5·227	3	3	10	2½	8·295
2	7¼	8	2⅛	5·326	3	3¼	10	3¼	8·402
2	7½	8	2⅞	5·411	3	3½	10	4	8·509
2	7¾	8	3¾	5·498	3	3¾	10	4⅞	8·617
2	8	8	4½	5·585	3	4	10	5⅝	8·726
2	8¼	8	5⅜	5·672	3	4¼	10	6⅜	8·836

TABLE 12 (*Continued*).

Diameter in feet & ins.		Circumference in feet & ins		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
3	4½	10	7¼	8·946	4	0½	12	8⅜	12·829
3	4¾	10	8	9·056	4	0¾	12	9⅛	12·962
3	5	10	8¾	9·168	4	1	12	9⅞	13·095
3	5¼	10	9½	9·211	4	1¼	12	10¾	13·229
3	5½	10	10⅜	9·393	4	1½	12	11½	13·364
3	5¾	10	11⅛	9·506	4	1¾	13	0¼	13·499
3	6	10	11⅞	9·621	4	2	13	1	13·635
3	6¼	11	0¾	9·736	4	2¼	13	1⅞	13·772
3	6½	11	1½	9·851	4	2½	13	2⅝	13·909
3	6¾	11	2¼	9·967	4	2¾	13	3⅜	14·047
3	7	11	3	10·084	4	3	13	4¼	14·186
3	7¼	11	3⅞	10·202	4	3¼	13	5	14·325
3	7½	11	4⅝	10·320	4	3½	13	5¾	14·465
3	7¾	11	5⅜	10·439	4	3¾	13	6½	14·606
3	8	11	6¼	10·559	4	4	13	7⅜	14·748
3	8¼	11	7	10·679	4	4¼	13	8⅛	14·890
3	8½	11	7¾	10·800	4	4½	13	8⅞	15·033
3	8¾	11	8½	10·922	4	4¾	13	9¾	15·176
3	9	11	9⅜	11·044	4	5	13	10½	15·320
3	9¼	11	0⅛	11·167	4	5¼	13	11¼	15·465
3	9½	11	10⅞	11·291	4	5½	14	0	15·611
3	9¾	11	11¾	11·415	4	5¾	14	0⅞	15·757
3	10	12	0½	11·534	4	6	14	1⅝	15·904
3	10¼	12	1¼	11·666	4	6¼	14	2⅜	16·051
3	10½	12	2	11·793	4	6½	14	3¼	16·200
3	10¾	12	2⅞	11·920	4	6¾	14	4	16·349
3	11	12	3⅝	12·048	4	7	14	4¾	16·498
3	11¼	12	4⅜	12·176	4	7¼	14	5½	16·649
3	11½	12	5¼	12·305	4	7½	14	6⅜	16·800
3	11¾	12	6	12·435	4	7¾	14	7⅛	16·951
4	0	12	6¾	12·566	4	8	14	7⅞	17·010
4	0¼	12	7½	12·697	4	8¼	14	8⅝	17·256

TABLE 12 (*Continued*).

Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
4	8½	14	9½	17·411	5	4½	16	10⅝	22·621
4	8¾	14	10¼	17·565	5	4¾	16	11⅜	22·866
4	9	14	11	17·720	5	5	17	0⅛	23·043
4	9¼	14	11⅞	17·875	5	5¼	17	0⅞	23·221
4	9½	15	0⅝	18·032	5	5½	17	1¾	23·330
4	9¾	15	1⅜	18·190	5	5¾	17	2½	23·578
4	10	15	2¼	18·347	5	6	17	3⅜	23·758
4	10¼	15	2⅞	18·506	5	6¼	17	4⅛	23·938
4	10½	15	3¾	18·663	5	6½	17	4⅞	24·119
4	10¾	15	4½	18·825	5	6¾	17	5⅝	24·301
4	11	15	5¼	18·985	5	7	17	6½	24·483
4	11¼	15	6⅛	19·141	5	7¼	17	7¼	24·666
4	11½	15	6⅞	19·309	5	7½	17	8	24·850
4	11¾	15	7¾	19·471	5	7¾	17	8¾	25·034
5	0	15	8½	19·635	5	8	17	9⅝	25·220
5	0¼	15	9¼	19·798	5	8¼	17	10⅜	25·405
5	0½	15	10	19·958	5	8½	17	11⅛	25·592
5	0¾	15	10¾	20·129	5	8¾	17	11⅞	25·779
5	1	15	11⅝	20·295	5	9	18	0¾	25·964
5	1¼	16	0⅜	20·461	5	9¼	18	1½	26·155
5	1½	16	1¼	20·629	5	9½	18	2¼	26·344
5	1¾	16	1⅞	20·797	5	9¾	18	3⅛	26·534
5	2	16	2¾	20·965	5	10	18	3⅞	26·725
5	2¼	16	3½	21·132	5	10¼	18	4⅝	26·916
5	2½	16	4¼	21·305	5	10½	18	5½	27·108
5	2¾	16	5⅛	21·476	5	10¾	18	6¼	27·301
5	3	16	5⅞	21·647	5	11	18	7	27·494
5	3¼	16	6¼	21·819	5	11¼	18	7¾	27·688
5	3½	16	7½	21·992	5	11½	18	8⅝	27·883
5	3¾	16	8¼	22·152	5	11¾	18	9⅜	28·078
5	4	16	9	22·333	6	0	18	10⅛	28·274
5	4¼	16	9¾	22·515	6	0¼	18	10⅞	28·471

TABLE 12 (*Continued*).

Diameter in feet & ins.	Circumference in feet & ins.	Areas in feet.	Diameter in feet & ins.	Circumference in feet & ins.	Areas in feet.
6 0½	18 11¾	28·668	6 8½	21 0⅞	35·344
6 0¾	19 0½	28·866	6 8¾	21 1⅝	35·564
6 1	19 1¼	29·065	6 9	21 2⅜	35·784
6 1¼	19 2⅛	29·264	6 9¼	21 3¼	36·006
6 1½	19 2⅞	29·466	6 9½	21 4	36·227
6 1¾	19 3⅝	29·665	6 9¾	21 4¾	36·450
6 2	19 4½	29·867	6 10	21 5½	36·674
6 2¼	19 5¼	30·069	6 10¼	21 6⅜	36·897
6 2½	19 6	30·271	6 10½	21 7⅛	37·122
6 2¾	19 6¾	30·475	6 10¾	21 7⅞	37·347
6 8	19 7⅝	30·679	6 11	21 8¾	37·573
6 3¼	19 8⅜	30·884	6 11¼	21 9½	37·700
6 3½	19 9⅛	31·090	6 11½	21 10¼	38·027
6 3¾	19 9⅞	31·296	6 11¾	21 11	38·256
6 4	19 10¾	31·503	7 0	21 11⅞	38·484
6 4¼	19 11½	31·710	7 1	22 3	39·406
6 4½	20 0¼	31·919	7 2	22 6⅛	40·338
6 4¾	20 1⅛	32·114	7 3	22 9¼	41·282
			7 4	23 0⅜	42·236
6 5	20 1⅞	32·337	7 5	23 2⅛	43·202
6 5¼	20 2⅝	32·548	7 6	23 6¾	44·178
6 5½	20 3½	32·759	7 7	23 11	45·165
6 5¾	20 4¼	32·970	7 8	24 1⅛	46·163
			7 9	24 4⅛	47·173
6 6	20 5	33·183	7 10	24 7¼	48·192
6 6¼	20 5¾	33·396	7 11	24 10¾	49·223
6 6½	20 6½	33·619			
6 6¾	20 7⅜	33·824	8 0	25 1½	50·265
			8 1	25 4⅝	51·317
6 7	20 8⅛	34·039	8 2	25 7⅞	52·381
6 7¼	20 8⅞	34·255	8 3	25 11	53·456
6 7½	20 9¾	34·471	8 4	26 2⅛	54·541
6 7¾	20 10½	34·688	8 5	26 5¼	55·637
			8 6	26 8⅜	56·745
6 8	20 11¼	34·906	8 7	26 11½	57·862
6 8¼	21 0⅛	35·125	8 8	27 2¾	58·992

TABLE 12 (*Continued*).

Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
8	9	27	5¾	60·132	11	10	37	2¾	109·977
8	10	27	9	61·282	11	11	37	5¼	111·531
8	11	28	0⅛	62·444					
					12	0	37	8⅜	113·097
9	0	28	3¼	63·617	12	1	37	11½	114·673
9	1	28	6⅜	64·800	12	2	38	2⅝	116·260
9	2	28	9½	65·995	12	3	38	5¾	117·859
9	3	29	0⅝	67·200	12	4	38	8⅞	119·467
9	4	29	3¾	68·416	12	5	39	0	121·087
9	5	29	7	69·644	12	6	39	3¼	122·718
9	6	29	10⅛	70·882	12	7	39	6⅜	124·359
9	7	30	1¼	72·130	12	8	39	9½	126·012
9	8	30	4⅜	73·391	12	9	40	0⅝	127·676
9	9	30	7½	74·662	12	10	40	3¾	129·350
9	10	30	11⅝	75·943	12	11	40	6⅞	131·036
9	11	31	1¾	77·236					
					13	0	40	10	132·732
10	0	31	5	78·540	13	1	41	1⅛	134·439
10	1	31	8⅛	79·854	13	2	41	4⅜	136·157
10	2	31	11¼	81·179	13	3	41	7½	137·886
10	3	32	2⅜	82·516	13	4	41	10½	139·626
10	4	32	5½	83·862	13	5	42	1⅝	141·377
10	5	32	8⅝	85·221	13	6	42	4⅞	143·139
10	6	32	11¾	86·590	13	7	42	8	144·911
10	7	33	2⅞	87·969	13	8	42	11⅛	146·694
10	8	33	6⅛	89·360	13	9	43	2¼	148·489
10	9	33	9¼	90·762	13	10	43	5½	150·294
10	10	34	0⅜	92·174	13	11	43	8⅝	152·110
10	11	34	3½	93·598					
					14	0	43	11¾	153·938
11	0	34	6⅝	95·033	14	1	44	2⅞	155·775
11	1	34	9¾	96·478	14	2	44	6	157·625
11	2	35	0⅞	97·934	14	3	44	9⅛	159·485
11	3	35	4⅛	99·402	14	4	45	0¼	161·355
11	4	35	7¼	100·879	14	5	45	3½	163·237
11	5	35	10⅝	102·368	14	6	45	6⅝	165·130
11	6	36	1½	103·869	14	7	45	9¾	167·033
11	7	36	4½	105·379	14	8	46	0⅞	168·947
11	8	36	7¾	106·901	14	9	46	4	170·873
11	9	36	10⅞	108·434	14	10	46	7⅛	172·809

TABLE 12 (*Continued*).

Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
14	11	46	11¼	174·756	18	0	56	6½	254·469
					18	1	56	9⅝	256·830
15	0	47	1½	176·715	18	2	57	0⅞	259·203
15	1	47	4⅝	178·683	18	3	57	4	261·587
15	2	47	7¾	180·663	18	4	57	7⅛	263·980
15	3	47	10⅞	182·654	18	5	57	10¼	266·386
15	4	48	2½	184·655	18	6	58	1⅜	268·803
15	5	48	5⅛	186·668	18	7	58	4½	271·229
15	6	48	8¼	188·692	18	8	58	7⅝	273·667
15	7	48	11⅜	190·726	18	9	58	10¾	276·117
15	8	49	2⅝	192·771	18	10	59	2	278·576
15	9	49	5¾	194·828	18	11	59	5⅛	281·047
15	10	49	8⅞	196·894					
15	11	50	0	198·973	19	0	59	8¼	283·528
					19	1	59	11⅛	285·972
16	0	50	3⅛	201·062	19	2	60	2⅝	288·472
16	1	50	6¼	203·161	19	3	60	5¾	290·972
16	2	50	9⅝	205·272	19	4	60	8⅞	293·564
16	3	51	0½	207·394	19	5	61	0	296·110
16	4	51	3¾	209·526	19	6	61	3⅛	298·648
16	5	51	6½	211·670	19	7	61	6¼	301·205
16	6	51	10	213·825	19	8	61	9⅜	303·774
16	7	52	1⅛	215·989	19	9	62	0½	306·355
16	8	52	4¼	218·166	19	10	62	3¾	308·945
16	9	52	7⅜	220·353	19	11	62	6⅞	311·547
16	10	52	10½	222·551					
16	11	53	1⅝	224·760	20	0	62	10	314·159
					20	1	63	1⅛	316·782
17	0	53	4⅞	226·980	20	2	63	4¼	319·417
17	1	53	8	229·210	20	3	63	7 7/16	322·062
17	2	53	11⅛	231·462	20	4	63	10 9/16	324·718
17	3	54	2⅛	233·705	20	5	64	1¾	327·385
17	4	54	5⅝	235·968	20	6	64	4⅞	330·063
17	5	54	8½	238·243	20	7	64	8	332·752
17	6	54	11⅝	240·528	20	8	64	11⅛	335·452
17	7	55	2⅞	242·824	20	9	65	2¼	338·163
17	8	55	6	245·131	20	10	65	5⅜	340·884
17	9	55	9⅛	247·450	20	11	65	8½	343·617
17	10	56	0¼	249·778					
17	11	56	3½	252·118	21	0	65	11⅜	346·360

TABLE 12 (*Continued*).

Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
21	1	66	2⅞	349·115	24	2	75	11	458·694
21	2	66	6	351·880	24	3	76	2¼	461·86
21	3	66	9⅛	354·656	24	4	76	5⅜	465·04
21	4	67	0¼	357·443	24	5	76	8½	468·23
21	5	67	3⅜	360·241	24	6	76	11⅝	471·435
21	6	67	6⅛	363·050	24	7	77	2¾	474·647
21	7	67	9⅝	365·870	24	8	77	5⅞	477·871
21	8	68	0⅞	368·700	24	9	77	9	481·105
21	9	68	4	371·542	24	10	78	0¼	484·35
21	10	68	7⅛	374·394	24	11	78	3⅜	487·61
21	11	68	10¼	377·258					
					25	0	78	6½	490·87
22	0	69	1⅝	380·132	25	1	78	9⅝	494·151
22	1	69	4½	383·018	25	2	79	0¾	497·44
22	2	69	7⅝	385·914	25	3	79	3⅞	500·74
22	3	69	10¾	388·821	25	4	79	7	504·051
22	4	70	1⅞	391·739	25	5	79	10⅛	507·37
22	5	70	5⅛	394·668	25	6	80	1⅜	510·705
22	6	70	8¼	397·607	25	7	80	4½	514·04
22	7	70	11⅜	400·558	25	8	80	7⅝	517·40
22	8	71	2½	403·520	25	9	80	10¾	520·76
22	9	71	5⅝	406·492	25	10	81	1⅞	524·144
22	10	71	8¾	409·476	25	11	81	5	527·531
22	11	72	0	412·070					
					26	0	81	8⅛	530·92
23	0	72	3⅛	415·475	26	1	81	11⅜	534·33
23	1	72	6¼	418·491	26	2	82	2½	537·757
23	2	72	9⅜	421·518	26	3	82	5⅝	541·18
23	3	73	0½	424·556	26	4	82	8¾	544·63
23	4	73	3⅝	427·605	26	5	82	11⅞	548·082
23	5	73	6¾	430·665	26	6	83	3	551·545
23	6	73	9⅞	433·736	26	7	83	6⅛	555·020
23	7	74	1⅛	436·817	26	8	83	9¼	558·505
23	8	74	4¼	439·910	26	9	84	0½	562·001
23	9	74	7⅜	443·013	26	10	84	3⅝	565·50
23	10	74	10½	446·128	26	11	84	6¾	569·026
23	11	75	1⅝	449·253					
					27	0	84	9⅞	572·555
24	0	75	4¾	452·389	27	1	85	1	576·095
24	1	75	7⅞	455·536	27	2	85	4⅛	579·645

TABLE 12 (*Continued*).

meter in & ins.	Circumference in feet & ins.		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
3	85	7¼	583·207	30	4	95	3½	722·653
4	85	10½	586·779	30	5	95	6⅝	726·629
5	86	1⅝	590·363	30	6	95	9⅞	730·616
6	86	4¾	593·957	30	7	96	1	734·614
7	86	7⅞	597·562	30	8	96	4⅛	738·623
8	86	11	601·178	30	9	96	7¼	742·643
9	87	2⅛	604·805	30	10	96	10⅜	746·673
10	87	5¼	608·443	30	11	97	1½	750·715
11	87	8¾	612·092					
				31	0	97	4⅝	754·767
0	87	11⅝	615·752	31	1	97	7⅞	758·831
1	88	2¾	619·422	31	2	97	11	762·905
2	88	5⅞	623·104	31	3	98	2⅛	766·904
3	88	9	626·796	31	4	98	5¼	771·086
4	89	0⅛	630·500	31	5	98	8⅜	775·193
5	89	3¼	634·214	31	6	98	11½	779·311
6	89	6⅜	637·939	31	7	99	2⅝	783·440
7	89	9⅝	641·675	31	8	99	5¾	787·579
8	90	0¾	645·422	31	9	99	9	791·730
9	90	3⅞	649·180	31	10	100	0⅛	795·892
10	90	7	652·949	31	11	100	3¼	800·064
11	90	10⅛	656·729					
				32	0	100	6⅜	804·247
0	91	1¼	660·519	32	1	100	9½	808·442
1	91	4⅜	664·321	32	2	101	0⅝	812·647
2	91	7½	668·133	32	3	101	3¾	816·863
3	91	10¾	671·957	32	4	101	7	821·090
4	92	1⅞	675·791	32	5	101	10⅛	825·328
5	92	5	679·636	32	6	102	1¼	829·576
6	92	8⅛	683·492	32	7	102	4⅜	833·836
7	92	11¼	687·359	32	8	102	7½	838·107
8	93	2⅜	691·237	32	9	102	10⅝	842·388
9	93	5½	695·126	32	10	103	1¾	846·681
10	93	8¾	699·026	32	11	103	4⅞	850·984
11	93	11⅞	702·936					
				33	0	103	8⅛	855·298
0	94	3	706·858	33	1	103	11¼	859·623
1	94	6⅛	710·790	33	2	104	2⅜	863·959
2	94	9¼	714·734	33	8	104	5½	868·306
3	95	0⅜	718·688	33	4	104	8⅝	872·664

TABLE 12 (*Continued*).

Diameter in feet & ins.	Circum-ference in feet & ins.	Areas in feet.	Diameter in feet & ins.	Circum-ference in feet & ins.	Areas in feet.
33 5	104 11¾	877·033	36 6	114 8	1046·346
33 6	105 2⅞	881·413	36 7	114 11⅛	1051·130
33 7	105 6	885·803	36 8	115 2¼	1055·924
33 8	105 9¼	890·205	36 9	115 5½	1060·729
33 9	106 0⅜	894·617	36 10	115 8⅝	1065·545
33 10	106 3½	899·040	36 11	115 11¾	1070·372
33 11	106 6⅝	903·475			
			37 0	116 2⅞	1075·210
34 0	106 9¾	907·920	37 1	116 6	1080·058
34 1	107 0⅞	912·376	37 2	116 9⅛	1084·918
34 2	107 4	916·343	37 3	117 0¼	1089·789
34 3	107 7¼	921·321	37 4	117 3⅜	1094·670
34 4	107 10⅜	925·869	37 5	117 6⅝	1099·562
34 5	108 1½	930·309	37 6	117 9¾	1104·466
34 6	108 4⅝	934·820	37 7	118 0⅞	1109·380
34 7	108 7¾	939·341	37 8	118 4	1114·305
34 8	108 10⅞	943·874	37 9	118 7⅛	1119·241
34 9	109 2	948·417	37 10	118 10¼	1124·188
34 10	109 5⅛	952·971	37 11	119 1⅜	1129·146
34 11	109 8⅜	957·536			
			38 0	119 4⅝	1134·114
35 0	109 11½	962·112	38 1	119 7¾	1139·094
35 1	110 2⅝	966·699	38 2	119 10⅞	1144·085
35 2	110 5¾	971·297	38 3	120 2	1149·086
35 3	110 8⅞	975·906	38 4	120 5⅛	1154·099
35 4	111 0	980·526	38 5	120 8¼	1159·122
35 5	111 3⅛	985·156	38 6	120 11⅜	1164·156
35 6	111 6⅜	989·798	38 7	121 2½	1169·201
35 7	111 9½	994·450	38 8	121 5¾	1174·257
35 8	112 0⅝	999·113	38 9	121 8⅞	1179·324
35 9	112 3¾	1003·787	38 10	122 0	1184·402
35 10	112 6⅞	1008·473	38 11	122 3⅛	1189·491
35 11	112 10	1013·169			
			39 0	122 6¼	1194·590
36 0	113 1⅛	1017·876	39 1	122 9⅜	1199·701
36 1	113 4¼	1022·593	39 2	123 0½	1204·822
36 2	113 7½	1027·322	39 3	123 3¾	1209·955
36 3	113 10⅝	1032·062	39 4	123 6⅞	1215·098
36 4	114 1¾	1036·812	39 5	123 10	1220·252
36 5	114 4⅞	1041·574	39 6	124 1⅛	1225·417

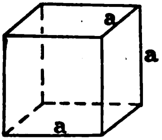
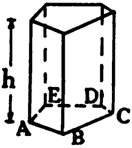
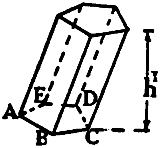
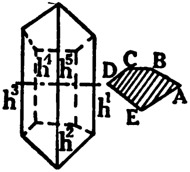
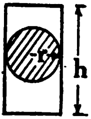
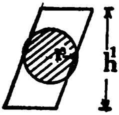
TABLE 12 (*Continued*).

Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
39	7	124	4¼	1230·593	42	8	134	0½	1429·773
39	8	124	7⅜	1235·780	42	9	134	3⅝	1435·364
39	9	124	10½	1240·978	42	10	134	6¾	1440·965
39	10	125	1⅝	1246·186	42	11	134	9⅞	1446·578
39	11	125	4⅞	1251·406					
					43	0	135	1	1452·201
40	0	125	8	1256·637	43	1	135	4¼	1457·835
40	1	125	11⅛	1261·878	43	2	135	7⅜	1463·480
40	2	126	2¼	1267·130	43	3	135	10½	1469·136
40	3	126	5⅜	1272·394	43	4	136	1⅝	1474·803
40	4	126	8½	1277·668	43	5	136	4¾	1480·481
40	5	126	11⅝	1282·953	43	6	136	7⅞	1486·169
40	6	127	2⅞	1288·249	43	7	136	11	1491·869
40	7	127	6	1293·556	43	8	137	2¼	1497·579
40	8	127	9⅛	1298·874	43	9	137	5⅜	1503·301
40	9	128	0¼	1304·202	43	10	137	8½	1509·033
40	10	128	3⅜	1309·542	43	11	137	11⅝	1514·776
40	11	128	6½	1314·892					
					44	0	138	2¾	1520·530
41	0	128	9⅝	1320·254	44	1	138	5⅞	1526·295
41	1	129	0¾	1325·626	44	2	138	9	1532·071
41	2	129	4	1331·009	44	3	139	0⅛	1537·858
41	3	129	7⅛	1336·404	44	4	139	3⅜	1543·656
41	4	129	10¼	1341·809	44	5	139	6½	1549·465
41	5	130	1⅜	1347·225	44	6	139	9⅝	1555·284
41	6	130	4½	1352·652	44	7	140	0¾	1561·115
41	7	130	7⅝	1358·089	44	8	140	3⅞	1566·956
41	8	130	10¾	1363·538	44	9	140	7	1572·808
41	9	131	2	1368·998	44	10	140	10⅛	1578·672
41	10	131	5⅛	1374·468	44	11	141	1⅜	1584·546
41	11	131	8¼	1379·950					
					45	0	141	4½	1590·431
42	0	131	11⅜	1385·442	45	1	141	7⅝	1596·327
42	1	132	2½	1390·945	45	2	141	10¾	1602·234
42	2	132	5⅝	1396·459	45	3	142	1⅞	1608·151
42	3	132	8¾	1401·984	45	4	142	5	1614·080
42	4	132	11⅞	1407·520	45	5	142	8⅛	1620·020
42	5	133	3⅛	1413·067	45	6	142	11¼	1625·970
42	6	133	6¼	1418·625	45	7	143	2½	1631·931
42	7	133	9⅜	1424·194	45	8	143	5⅝	1637·904

TABLE 12 (*Continued*).

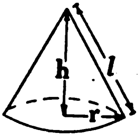
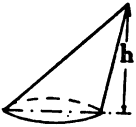
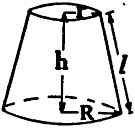
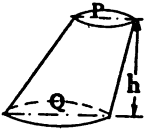
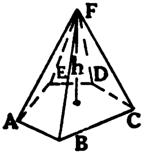
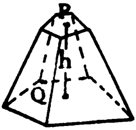
Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.	Diameter in feet & ins.		Circumference in feet & ins.		Areas in feet.
45	9	143	8¾	1643·887	48	0	150	9½	1809·557
45	10	143	11⅞	1649·881	48	1	151	0¾	1815·846
45	11	144	3	1655·881	48	2	151	3⅞	1822·145
					48	3	151	7	1828·456
46	0	144	6⅛	1661·902	48	4	151	10⅛	1834·777
46	1	144	9¼	1667·929	48	5	152	1¼	1841·109
46	2	145	0½	1673·967	48	6	152	4⅜	1847·452
46	3	145	3⅝	1680·015	48	7	152	7½	1853·806
46	4	145	6¾	1686·075	48	8	152	10¾	1860·171
46	5	145	9⅞	1692·145	48	9	153	1⅞	1866·547
46	6	146	1	1698·227	48	10	153	5	1872·934
46	7	146	4⅛	1704·319	48	11	153	8⅛	1879·332
46	8	146	7¼	1710·422					
46	9	146	10⅜	1716·536	49	0	153	11¼	1885·741
46	10	147	1⅝	1722·661	49	1	154	2⅜	1892·160
46	11	147	4¾	1728·797	49	2	154	5½	1898·591
					49	3	154	8⅝	1905·032
47	0	147	7⅞	1734·944	49	4	154	11⅞	1911·484
47	1	147	11	1741·102	49	5	155	3	1917·947
47	2	148	2⅛	1747·270	49	6	155	6⅛	1924·421
47	3	148	5¼	1753·450	49	7	155	9¼	1930·906
47	4	148	8⅜	1759·641	49	8	156	0⅜	1937·402
47	5	148	11⅜	1765·842	49	9	156	3½	1943·909
47	6	149	2¾	1772·054	49	10	156	6⅝	1950·427
47	7	149	5⅞	1778·277	49	11	156	9⅞	1956·955
47	8	149	9	1784·511					
47	9	150	0⅛	1790·756	50	0	157	1	1963·495
47	10	150	3¼	1797·012					
47	11	150	6⅜	1803·279					

MENSURATION

FORM.	NAME.
	Cube
	Right Parallelopiped
	Oblique Parallelopiped (with parallel end surfaces)
	Truncated Parallelopiped, with n parallel edges of length— $h^1, h^2, h^3 \ldots h^n$. Section perpendicular to the edges h^1, h^2, etc.
	Right Cylinder (with circular base)
	Oblique Cylinder (with circular base)

OF SOLIDS

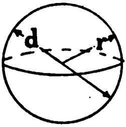
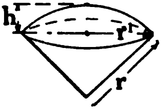
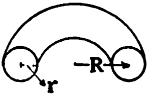
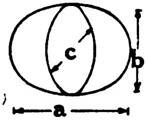
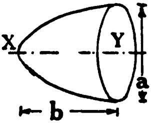
AREA OF SURFACE (S).	VOLUME (V).
$S = 6a^2$	$V = a^3$
Sides $= \left\{ h \, (AB + BC + CD + DE + EA) \right.$ Total $=$ Sides $+ 2 \, (ABCDE)$	$V = h \times$ area ABCDE
h^1 = perpendicular between the two ends Sides $= \left\{ h^1 \, (AB + BC + CD + DE + EA) \right.$ Total $=$ Sides $+ 2 \, (ABCDE)$	$V = h^1 \times$ area ABCDE
Sides $=$ $\frac{AB}{2} (h^1 + h^2) + \frac{BC}{2} (h^2 + h^3)$ $+ \frac{CD}{2} (h^3 + h^4) + \frac{DE}{2} (h^4 + h^5)$ $+ \frac{EA}{2} (h^5 + h^1)$ Total $=$ Sides $+ 2$ end polygons	$V =$ Area ABCD $\times \left(\frac{h^1 + h^2 + h^3 \ldots h^n}{n} \right)$
Curved surface $= 2\pi rh$ Total surface $= 2\pi rh + 2\pi r^2$	$V = \pi r^2 h$
Curved surface $= l \times 2\pi r$	$V = \pi r^2 h^1$

Form.	Name.
	Right Cone (circular base)
	Oblique Cone
	Frustum of Cone
	Truncated Oblique Cone (with parallel bases of area P and Q)
	Pyramid
	Truncated Pyramid (parallel bases of area P and Q)

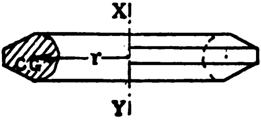
OF SOLIDS

AREA OF SURFACE (S).	VOLUME (V).
Area curved surface— $\begin{cases} = \pi r l \\ = \pi r\sqrt{r^2+h^2} \end{cases}$ Total S $= \frac{5}{8}r^2+\frac{5}{8}r\sqrt{r^2+h^2}$	$V=\frac{\pi}{3}r^2h$
——	$V=\frac{1}{3}$ ((area base) h where h is height of apex above the plane of the base
Area curved surface— $=\pi(R+r)\sqrt{(R-r)^2+h^2}$ *Note* $l=\sqrt{(R-r)^2+h^2}$ Total S $=\pi R^2+\pi r^2+$ area curved surface	$V=\frac{\pi h}{3}(R^2+r^2+Rr)$
——	$V=\frac{h}{3}(P+Q+\sqrt{PQ})$
S = Area of base ABCDE + area of triangles ABF, BCF, CDF, DEF, EAF	$V=\frac{1}{3}h$ (area base ABCDE)
——	$V=\frac{h}{3}(P+Q+\sqrt{PQ})$

MENSURATION

Form.	Name.
	Sphere
	Sector of Sphere
	Ring
	Ellipsoid of three axes *a, b, c*
	Paraboloid of Revolution (about axes XY)

Gulden's Theorem.



Volume obtained by the rotation of a plane surface about an axis situated in the same plane.

If S is the area of the surface and r is the distance of its Centre of Gravity from the axis XY, then the Volume $V = S \times 2\pi r$.

OF SOLIDS

AREA OF SURFACE (S).	VOLUME (V).
$S = \pi d^2$ $S = 4\pi r^2$ $= 12{\cdot}56 \times r^2$	$= \frac{1}{6}\pi d^3 = {\cdot}5236 \times d^3$ $V = \frac{4}{3}\pi r^3 = 4{\cdot}189 \times r^3$
$S = \pi r\,(2h + r^1)$	$V = \frac{2}{3}\pi r^2 h$ $= 2{\cdot}094 r^2 h$
$S = (2\pi r) \times (2\pi R)$ $= 4\pi^2 R r$	$V = 2\pi^2 r^2 R$
——	$V = \frac{1}{6}\pi abc$
——	$V = \frac{\pi}{8} a^2 b$

TABLE 13.

DECIMAL EQUIVALENTS OF FRACTIONS.

Fractions.	Decimals.	Fractions.	Decimals.	Fractions.	Decimals.	Fractions.	Decimals.
$\frac{1}{64}$	0·015625	$\frac{17}{64}$	0·265625	$\frac{33}{64}$	0·515625	$\frac{49}{64}$	0·765625
$\frac{1}{32}$	0·03125	$\frac{9}{32}$	0·28125	$\frac{17}{32}$	0·53125	$\frac{25}{32}$	0·78125
$\frac{3}{64}$	0·046875	$\frac{19}{64}$	0·296875	$\frac{35}{64}$	0·546875	$\frac{51}{64}$	0·796875
$\frac{1}{16}$	0·0625	$\frac{5}{16}$	0·3125	$\frac{9}{16}$	0·5625	$\frac{13}{16}$	0·8125
$\frac{5}{64}$	0·078125	$\frac{21}{64}$	0·328125	$\frac{37}{64}$	0·578125	$\frac{53}{64}$	0·828125
$\frac{3}{32}$	0·09375	$\frac{11}{32}$	0·34375	$\frac{19}{32}$	0·59375	$\frac{27}{32}$	0·84375
$\frac{7}{64}$	0·109375	$\frac{23}{64}$	0·359375	$\frac{39}{64}$	0·609375	$\frac{55}{64}$	0·859375
$\frac{1}{8}$	0·125	$\frac{3}{8}$	0·375	$\frac{5}{8}$	0·625	$\frac{7}{8}$	0·875
$\frac{9}{64}$	0·140625	$\frac{25}{64}$	0·390625	$\frac{41}{64}$	0·640625	$\frac{57}{64}$	0·890625
$\frac{5}{32}$	0·15625	$\frac{13}{32}$	0·40625	$\frac{21}{32}$	0·65625	$\frac{29}{32}$	0·90625
$\frac{11}{64}$	0·171875	$\frac{27}{64}$	0·421875	$\frac{43}{64}$	0·671875	$\frac{59}{64}$	0·921875
$\frac{3}{16}$	0·1875	$\frac{7}{16}$	0·4375	$\frac{11}{16}$	0·6875	$\frac{15}{16}$	0·9375
$\frac{13}{64}$	0·203125	$\frac{29}{64}$	0·453125	$\frac{45}{64}$	0·703125	$\frac{61}{64}$	0·953125
$\frac{7}{32}$	0·21875	$\frac{15}{32}$	0·46875	$\frac{23}{32}$	0·71875	$\frac{31}{32}$	0·96875
$\frac{15}{64}$	0·234375	$\frac{31}{64}$	0·484375	$\frac{47}{64}$	0·734375	$\frac{63}{64}$	0·984375
$\frac{1}{4}$	0·25	$\frac{1}{2}$	0·5	$\frac{3}{4}$	0·75	1	1·0

TABLE 14.

INCHES AND PARTS OF AN INCH IN DECIMALS OF A FOOT.

Part of an inch.		1	2	3	4	5	6	7	8	9	10	11
		·0833	·1666	·2500	·3333	·4166	·5000	·5833	·6666	·7500	·8333	·9166
1/32	·0026	·0859	·1692	·2526	·3359	·4192	·5026	·5859	·6692	·7526	·8359	·9192
1/16	·0052	·0885	·1718	·2552	·3385	·4218	·5052	·5885	·6718	·7552	·8385	·9218
3/32	·0078	·0911	·1744	·2578	·3411	·4244	·5078	·5911	·6744	·7578	·8411	·9240
1/8	·0104	·0937	·1770	·2604	·3437	·4270	·5104	·5937	·6770	·7604	·8437	·9270
5/32	·0130	·0963	·1796	·2630	·3463	·4296	·5130	·5963	·6796	·7630	·8463	·9296
3/16	·0156	·0989	·1822	·2656	·3489	·4322	·5156	·5989	·6822	·7656	·8489	·9322
7/32	·0182	·1015	·1848	·2682	·3515	·4348	·5182	·6015	·6848	·7682	·8515	·9348
1/4	·0208	·1041	·1875	·2708	·3541	·4374	·5208	·6041	·6874	·7708	·8541	·9374
9/32	·0234	·1067	·1901	·2734	·3567	·4401	·5234	·6067	·6901	·7734	·8567	·9401
5/16	·0260	·1093	·1927	·2760	·3593	·4427	·5260	·6093	·6927	·7760	·8593	·9427
11/32	·0286	·1119	·1953	·2786	·3619	·4453	·5286	·6119	·6953	·7786	·8619	·9453
3/8	·0312	·1145	·1979	·2812	·3645	·4479	·5312	·6145	·6979	·7812	·8645	·9479
13/32	·0338	·1171	·2005	·2838	·3671	·4505	·5338	·6171	·7005	·7838	·8671	·9505
7/16	·0364	·1197	·2031	·2864	·3697	·4531	·5364	·6197	·7031	·7864	·8697	·9531
15/32	·0390	·1223	·2057	·2890	·3723	·4557	·5390	·6223	·7057	·7890	·8723	·9557
1/2	·0416	·1249	·2083	·2916	·3749	·4583	·5416	·6249	·7083	·7916	·8749	·9583
17/32	·0442	·1276	·2109	·2942	·3776	·4609	·5442	·6276	·7109	·7942	·8776	·9609
9/16	·0468	·1302	·2135	·2968	·3802	·4635	·5468	·6302	·7135	·7968	·8802	·9635
19/32	·0494	·1328	·2161	·2994	·3828	·4661	·5494	·6328	·7161	·7994	·8828	·9661
5/8	·0520	·1354	·2187	·3020	·3854	·4687	·5520	·6354	·7187	·8020	·8854	·9687
21/32	·0546	·1380	·2213	·3046	·3880	·4713	·5546	·6380	·7213	·8046	·8880	·9713
11/16	·0572	·1406	·2239	·3072	·3906	·4739	·5572	·6406	·7239	·8072	·8906	·9739
23/32	·0598	·1432	·2265	·3098	·3932	·4765	·5598	·6432	·7265	·8098	·8932	·9765
3/4	·0625	·1458	·2291	·3124	·3958	·4791	·5625	·6458	·7291	·8124	·8958	·9791
25/32	·0651	·1484	·2317	·3151	·3984	·4817	·5651	·6484	·7317	·8151	·8984	·9817
13/16	·0677	·1510	·2343	·3177	·4010	·4843	·5677	·6510	·7343	·8177	·9010	·9843
27/32	·0702	·1536	·2369	·3203	·4036	·4869	·5703	·6536	·7369	·8203	·9036	·9869
7/8	·0729	·1562	·2395	·3229	·4062	·4895	·5729	·6562	·7395	·8229	·9062	·9895
29/32	·0755	·1588	·2421	·3255	·4088	·4921	·5755	·6588	·7421	·8255	·9088	·9921
15/16	·0787	·1614	·2447	·3281	·4114	·4947	·5781	·6614	·7447	·8281	·9114	·9947
31/32	·0801	·1640	·2473	·3307	·4140	·4973	·5807	·6640	·7473	·8307	·9140	·9973

Example. 1/2″ = ·0416ft. 5 7/8″ = ·4895ft.

Useful Constants for Conversion.

Inches	×	25·399	=	Millimetres.
Millimetres	×	·03937	=	Inches.
Feet	×	·3048	=	Metres.
Yards	×	·9144	=	Metres.
Metres	×	39·37	=	Inches.
Metres	×	3·2808	=	Feet.
Metres	×	1·0936	=	Yards.
Kilometres	×	·6214	=	Miles.
Miles	×	1·6093	=	Kilometres.
Square Centimetres	×	·1550	=	Square Inches.
Square Inches	×	6·4513	=	Square Centimetres.
Square Metres	×	10·7643	=	Square Feet.
Square Metres	×	1·196	=	Square Yards.
Square Yards	×	·836	=	Square Metres.
Square Feet	×	·0929	=	Square Metres.
Square Miles	×	2·5899	=	Square Kilometres.
Acres	×	·4047	=	Hectares.
Ares	×	·0247	=	Acres.
Cubic Centimetres	×	·0610	=	Cubic Inches.
Cubic Inches	×	16·3862	=	Cubic Centimetres.
Cubic Metres	×	35·3166	=	Cubic Feet.
Cubic Feet	×	·0283	=	Cubic Metres.
Centimetres^4	×	·024025	=	Inches^4.
Inches^4	×	41·616	=	Centimetres^4.
Grammes	×	15·44	=	Grains.
Grains	×	·0648	=	Grammes.
Kilogrammes	×	2·2046	=	Pounds.
Pounds	×	·4535	=	Kilogrammes.
1 metric ton (1,000 Kilogrammes)	×	·9842	=	Tons.
Litres	×	61·023	=	Cubic Inches.
Litres	×	·2201	=	Gallons.
Gallons	×	4·5437	=	Litres.
Litres	×	1·761	=	Pints.
Pints	×	·568	=	Litres.
Kilogrammes	×	7·233	=	Foot-pounds.
Foot-pounds	×	·13825	=	Kilogrammetres.
Force de Cheval	×	1·0138	=	Horse Power.
Horse Power	×	·9863	=	Force de Cheval.
Litres per Force de Cheval	×	1·785	=	Pints per Horse Power.
Pints per Horse Power	×	·5601	=	Litres per Force de Cheval.
Kilogrammes per Sq. Metre	×	·2048	=	Pounds per Square Foot.
Pounds per Sq. Foot	×	4·882	=	Kilogrammes per Sq. Metre.

Pounds per Sq. Inch . × ·0703 = Kilogrammes per Sq. Centimetre.

Grammes per Sq. Metre × ·0294912 = Ounces per Sq. Yard.

Ounces per Sq. Yard . × 33·9239 = Grammes per Sq. Metre.

Kilogrammes per Sq. Centimetre . . 14·223 = Pounds per Sq. Inch.

Pounds per Sq. Inch . × ·0703 = Kilogrammes per Sq. Centimetre.

Kilogrammes per Cubic Centimetre . . × 36·1253 = Pounds per Cub. In.

Kilogrammes per Cubic Metre . . . × ·0624245 = Pounds per Cub. Ft.

Kilogrammes per Cubic Metre . . . × 1·68546 = Pounds per Cub. Yd.

Pounds per Cubic Inch × ·02728 = Kilogrammes per Cub. Centimetre.

Pounds per Cubic Foot × 16·019 = Kilogrammes per Cub. Metre.

Pounds per Cubic Yard × ·5933 = Kilogrammes per Cub. Metre.

Kilogrammes per Metre run . . . × 2·016 = Pounds per Yard.

Kilogrammes per Metre run . . . × ·672 = Pounds per Foot.

Pounds per Yard run . × ·496 = Kilogrammes per Metre.

Pounds per Foot run . × 1·488 = Kilogrammes per Metre.

British Thermal Heat Units . . . × ·252 = Kilogramme Calories

Metres per second . × 3·281 = Feet per Second.

Feet per Second . . × ·3048 = Metres per Second.

Metres per Second . × 2·237 = Miles per Hour.

Miles per hour . . × ·447 = Metres per Second.

$\frac{9}{5}$ (Degrees Centigrade) + 32 = Degrees Fahrenheit.

$\frac{5}{9}$ (Degrees Fahrenheit. − 32) = Degrees Centigrade.

1 Kilo. watt = 1,000 Watts = 1·346 horse-power.

1 Horse-Power = 550 ft. lbs. per sec. = 746 Watts

1 Imperial Gallon of Water, weighs 10 lbs. = 4·5437 litres.

1 Cu. Foot of Water weighs 62·35 lbs. = 28·378 litres.

Pressure due to 1 Atmosphere
= 14·7 lbs. per sq. inch.
= 1·0335 kgms. per sq. cm.
= 760 mm. of Mercury.
= 29·922 ins. Mercury.
= 34·947 feet of water.

Head of water in feet × ·4325 = pressure in lbs. sq. in. (fresh water).

1-inch Water gauge = ·03604 lbs. per sq. inch.

1-inch Mercury gauge = ·49116 lbs. per sq. in.
Inches of Rainfall × 100·925 = tons per acre.
Weight of Snow = ·433 lb. per sq. inch per inch depth.
Admiralty Knot = Speed of 6.080 feet per hour.
Feet per minute × ·01136 = miles per hour.
Miles per hour × ·86742 = knots.
Wrought iron cub. inches × ·28 = weight in lbs.
1 stone = 14 lbs.
1 dozen = 12
1 gross = 12 dozen.
1 ream = 20 quires.
1 quire = 24 sheets.
1 score = 20.
1 Imperial Drawing Board = 30 ins. × 22 ins.
1 Double Elephant Drawing Board = 40 ins. × 27 ins.
1 mil = ·001 inch.
1 fathom = 6 feet.
1 link = 7·92 ins
1 sq. mile = 640 acres.
1 circular inch = area of circle of 1-inch diameter = ·7854 sq. inches.

Archine (Turkish) = 0·68 sq. m.
Bekre (Turkish) = 20 deunums.
Berkovets = 400 Russ. lbs.
Cent. = Amer. ½d.; Dutch ⅕d.
Centavo (Portuguese) = ½d.
Centime . = ⅒d.
Centner (German) = 1 cwt.
Conto = Brazil £50; Portug. £200.
Copeck (Russ.) = ¼d.
Decagram = ½ oz.
Dekar . = 1,200 sq. yds.
Desyatina (Russ.) = 2·7 acres.
Deunum (Turkish) = ¼ acre.
Drachma (Greek) = 9½d.
Dramme (Turkish) = ·112 ozs.
Drittel, Drittlar (Scand.) = 110 lbs.
Escudo (Portuguese) = 4s. 6d.
Filler (Hungarian) = ⅒d.
Florin, Guilder (Dutch) = 1s. 8d.
Franc . . = 9½d.
Gramme . . = ·035 oz.
Grosh (Turkish) = 2d.
Heller (Austrian) ⅒d.
Joch, Katasterjoch (Austr.) = 1⅖ acres.
Krone = Scand. 1s. 1⅓d., Austria 10d.
Lepton (Greek) = ⅒d.
Leu, lev, lire . = 9½d.
Logel (German) = 50 litres.
Mark (German) = 11¾d.
Mazsa (Hungarian) = 1cwt.
Metercentner, metermazsa . = 2 cwt.
Milreis (Portuguese) . . = 4s. 6d.
Morgen (Prussia) = ⅝ acre.
Oka (Turkish) . = 3 lbs.
Ore (Scand.) . = ⅐d.
Para (Turkish) = $\frac{1}{20}$d.
Peseta (Spanish) = 9½d.
Pfennig (German) = ⅛d.
Piastre (Turkish) = 2d.
Pud (Russian) = 36 lbs.
Quintal (metric) = 2 cwt.
,, sometimes 100 lbs.
Ratal (Turkish) = 2 okas.
Real (Portuguese) ·054d.
Rouble (Russ.) = 2s. 1½d.

Sagene (Russ.) = 7 ft.
Skippund (Norwegian) . = 350 lbs.
Skjoeppe (Norwegian) . ½ bushel.
Vedro (Russ.) = 2·7 gall.
Vershok (Russ.) = 1¾ inch.
Verst (Russ.) = ⅔ mile.
Yoke, Cadastral (Austr.) . = 1⅜ acres.
Zolotnik (Russ.) = 0·15 oz.

Note.—All coinage equivalents on pre-war exchange rates.

TABLE 15.

EQUIVALENTS OF MILLIMETRES IN INCHES
Advancing by Decimals of an Inch.

M/m	Ins.	M/m	Ins.	M/m	Ins.	M/m	Ins.	M/m	Ins.
1	·039	51	2·008	101	3·976	151	5·945	201	7·913
2	·079	52	2·047	102	4·016	152	5·984	202	7·963
3	·118	53	2·087	103	4·055	153	6·024	203	7·992
4	·157	54	2·126	104	4·095	154	6·063	204	8·032
5	·197	55	2·165	105	4·134	155	6·102	205	8·071
6	·236	56	2·205	106	4·173	156	6·142	206	8·110
7	·276	57	2·244	107	4·213	157	6·181	207	8·150
8	·315	58	2·283	108	4·252	158	6·221	208	8·189
9	·354	59	2·323	109	4·291	159	6·260	209	8·228
10	·394	60	2·362	110	4·331	160	6·299	210	8·268
11	·433	61	2·402	111	4·370	161	6·339	211	8·307
12	·472	62	2·441	112	4·409	162	6·378	212	8·347
13	·512	63	2·480	113	4·449	163	6·417	213	8·386
14	·551	64	2·520	114	4·488	164	6·457	214	8·425
15	·591	65	2·559	115	4·528	165	6·496	215	8·465
16	·630	66	2·598	116	4·567	166	6·535	216	8·504
17	·669	67	2·638	117	4·606	167	6·575	217	8·543
18	·709	68	2·677	118	4·646	168	6·614	218	8·583
19	·748	69	2·717	119	4·685	169	6·654	219	8·622
20	·787	70	2·756	120	4·724	170	6·693	220	8·661
21	·827	71	2·795	121	4·764	171	6·732	221	8·701
22	·866	72	2·835	122	4·803	172	6·772	222	8·740
23	·906	73	2·874	123	4·843	173	6·811	223	8·780
24	·945	74	2·913	124	4·882	174	6·850	224	8·819
25	·984	75	2·953	125	4·921	175	6·890	225	8·858
26	1·024	76	2·992	126	4·961	176	6·929	226	8·898
27	1·063	77	3·032	127	5·000	177	6·969	227	8·937
28	1·102	78	3·071	128	5·039	178	7·008	228	8·976
29	1·142	79	3·110	129	5·079	179	7·047	229	9·016
30	1·181	80	3·150	130	5·118	180	7·087	230	9·055
31	1·220	81	3·189	131	5·158	181	7·126	231	9·095
32	1·260	82	3·228	132	5·197	182	7·165	232	9·134
33	1·299	83	3·268	133	5·236	183	7·205	233	9·173
34	1·339	84	3·307	134	5·276	184	7·244	234	9·213
35	1·378	85	3·346	135	5·315	185	7·284	235	9·252
36	1·417	86	3·386	136	5·354	186	7·323	236	9·291
37	1·457	87	3·425	137	5·394	187	7·362	237	9·331
38	1·496	88	3·465	138	5·433	188	7·402	238	9·370
39	1·535	89	3·504	139	5·472	189	7·441	239	9·410
40	1·575	90	3·543	140	5·512	190	7·480	240	9·449
41	1·614	91	3·583	141	5·551	191	7·520	241	9·488
42	1·654	92	3·622	142	5·591	192	7·559	242	9·528
43	1·693	93	3·661	143	5·630	193	7·598	243	9·567
44	1·732	94	3·701	144	5·669	194	7·638	244	9·606
45	1·772	95	3·740	145	5·709	195	7·677	245	9·646
46	1·811	96	3·780	146	5·748	196	7·717	246	9·685
47	1·850	97	3·819	147	5·787	197	7·756	247	9·724
48	1·890	98	3·858	148	5·827	198	7·795	248	9·764
49	1·929	99	3·898	149	5·866	199	7·835	249	9·803
50	1·969	100	3·937	150	5·906	200	7·874	250	9·843

TABLE 15 (*Continued*).

M/M	Ins.	M/m	ns.	M/m	Ins.	M/m	Ins.	M/m	Ins.
251	9·882	301	11·850	351	13·819	401	15·788	451	17·756
252	9·921	302	11·890	352	13·858	402	15·827	452	17·795
253	9·961	303	11·929	353	13·898	403	15·866	453	17·835
254	10·000	304	11·969	354	13·937	404	15·906	454	17·874
255	10·039	305	12·008	355	13·977	405	15·945	455	17·914
256	10·079	306	12·047	356	14·016	406	15·984	456	17·953
257	10·118	307	12·087	357	14·055	407	16·024	457	17·992
258	10·158	308	12·126	358	14·095	408	16·063	458	18·032
259	10·197	309	12·165	359	14·134	409	16·103	459	18·071
260	10·236	310	12·205	360	14·173	410	16·142	460	18·110
261	10·276	311	12·244	361	14·213	411	16·181	461	18·150
262	10·315	312	12·284	362	14·252	412	16·221	462	18·189
263	10·354	313	12·323	363	14·291	413	16·260	463	18·229
264	10·394	314	12·362	364	14·331	414	16·299	464	18·268
265	10·433	315	12·402	365	14·370	415	16·339	465	18·307
266	10·473	316	12·441	366	14·410	416	16·378	466	18·347
267	10·512	317	12·480	367	14·449	417	16·417	467	18·386
268	10·551	318	12·520	368	14·488	418	16·457	468	18·425
269	10·591	319	12·559	369	14·528	419	16·496	469	18·465
270	10·630	320	12·599	370	14·567	420	16·536	470	18·504
271	10·669	321	12·638	371	14·606	421	16·575	471	18·543
272	10·709	322	12·677	372	14·646	422	16·614	472	18·583
273	10·748	323	12·717	373	14·685	423	16·654	473	18·622
274	10·787	324	12·756	374	14·725	424	16·693	474	18·662
275	10·827	325	12·795	375	14·764	425	16·732	475	18·701
276	10·866	326	12·835	376	14·803	426	16·772	476	18·740
277	10·906	327	12·874	377	14·843	427	16·811	477	18·780
278	10·945	328	12·913	378	14·882	428	16·851	478	18·819
279	10·984	329	12·953	379	14·921	429	16·890	479	18·858
280	11·024	330	12·992	380	14·961	430	16·929	480	18·898
281	11·063	331	13·032	381	15·000	431	16·969	481	18·937
282	11·102	332	13·071	382	15·040	432	17·008	482	18·977
283	11·142	333	13·110	383	15·079	433	17·047	483	19·016
284	11·181	334	13·150	384	15·118	434	17·087	484	19·055
285	11·221	335	13·189	385	15·158	435	17·126	485	19·095
286	11·260	336	13·228	386	15·197	436	17·166	486	19·134
287	11·299	337	13·268	387	15·236	437	17·205	487	19·173
288	11·339	338	13·307	388	15·276	438	17·244	488	19·213
289	11·378	339	13·347	389	15·315	439	17·284	489	19·252
290	11·417	340	13·386	390	15·354	440	17·323	490	19·292
291	11·457	341	13·425	391	15·394	441	17·362	491	19·331
292	11·496	342	13·465	392	15·433	442	17·402	492	19·370
293	11·536	343	13·504	393	15·473	443	17·441	493	19·410
294	11·575	344	13·543	394	15·512	444	17·480	494	19·449
295	11·614	345	13·583	395	15·551	445	17·520	495	19·488
296	11·654	346	13·622	396	15·591	446	17·559	496	19·528
297	11·693	347	13·662	397	15·630	447	17·599	497	19·567
298	11·732	348	13·701	398	15·669	448	17·638	498	19·606
299	11·772	349	13·740	399	15·709	449	17·677	499	19·646
300	11·811	350	13·780	400	15·748	450	17·717	500	19·685

TABLE 15 (*Continued*).

M/m	Ins.	M/m	Ins.	M/m	Ins.	M/m	Ins.	M/m	Ins.
501	19·725	551	21·693	601	23·662	651	25·630	701	27·599
502	19·764	552	21·732	602	23·701	652	25·670	702	27·638
503	19·803	553	21·772	603	23·740	653	25·709	703	27·677
504	19·843	554	21·811	604	23·780	654	25·748	704	27·717
505	19·882	555	21·851	605	23·819	655	25·788	705	27·756
506	19·921	556	21·890	606	23·858	656	25·827	706	27·796
507	19·961	557	21·929	607	23·898	657	25·866	707	27·835
508	20·000	558	21·969	608	23·937	658	25·906	708	27·874
509	20·040	559	22·008	609	23·977	659	25·945	709	27·914
510	20·079	560	22·047	610	24·016	660	25·984	710	27·953
511	20·118	561	22·087	611	24·055	661	26·024	711	27·992
512	20·158	562	22·126	612	24·095	662	26·063	712	28·032
513	20·197	563	22·166	613	24·134	663	26·103	713	28·071
514	20·236	564	22·205	614	24·173	664	26·142	714	28·110
515	20·276	565	22·244	615	24·213	665	26·181	715	28·150
516	20·315	566	22·284	616	24·252	666	26·221	716	28·189
517	20·355	567	22·323	617	24·292	667	26·260	717	28·229
518	20·394	568	22·362	618	24·331	668	26·299	718	28·268
519	20·433	569	22·402	619	24·370	669	26·339	719	28·307
520	20·473	570	22·441	620	24·410	670	26·378	720	28·347
521	20·512	571	22·481	621	24·449	671	26·418	721	28·386
522	20·551	572	22·520	622	24·488	672	26·457	722	28·425
523	20·591	573	22·559	623	24·528	673	26·496	723	28·465
524	20·630	574	22·599	624	24·567	674	26·536	724	28·504
525	20·669	575	22·638	625	24·607	675	26·575	725	28·544
526	20·709	576	22·677	626	24·646	676	26·614	726	28·583
527	20·748	577	22·717	627	24·685	677	26·654	727	28·622
528	20·788	578	22·756	628	24·725	678	26·693	728	28·662
529	20·827	579	22·795	629	24·764	679	26·733	729	28·701
530	20·866	580	22·835	630	24·803	680	26·772	730	28·740
531	20·906	581	22·874	631	24·843	681	26·811	731	28·780
532	20·945	582	22·914	632	24·882	682	26·851	732	28·819
533	20·984	583	22·953	633	24·921	683	26·890	733	28·859
534	21·024	584	22·992	634	24·961	684	26·929	734	28·898
535	21·063	585	23·032	635	25·000	685	26·969	735	28·937
536	21·103	586	23·071	636	25·040	686	27·008	736	28·977
537	21·142	587	23·110	637	25·079	687	27·047	737	29·016
538	21·181	588	23·150	638	25·118	688	27·087	738	29·055
539	21·221	589	23·189	639	25·158	689	27·126	739	29·095
540	21·260	590	23·229	640	25·197	690	27·166	740	29·134
541	21·299	591	23·268	641	25·236	691	27·205	741	29·173
542	21·339	592	23·307	642	25·276	692	27·244	742	29·213
543	21·378	593	23·347	643	25·315	693	27·284	743	29·252
544	21·418	594	23·386	644	25·355	694	27·323	744	29·292
545	21·457	595	23·425	645	25·394	695	27·362	745	29·331
546	21·496	596	23·464	646	25·433	696	27·402	746	29·370
547	21·536	597	23·503	647	25·473	697	27·441	747	29·410
548	21·575	598	23·543	648	25·512	698	27·481	748	29·449
549	21·614	599	23·582	649	25·551	699	27·520	749	29·488
550	21·654	600	23·622	650	25·591	700	27·559	750	29·528

TABLE 15 (*Continued*).

M/m	Ins.	M/m	Ins.	M/m	Ins.	M/m	Ins.	M/m	Ins.
751	29·567	801	31·536	851	33·504	901	35·473	951	37·441
752	29·607	802	31·575	852	33·544	902	35·512	952	37·481
753	29·646	803	31·614	853	33·583	903	35·552	953	37·520
754	29·685	804	31·654	854	33·622	904	35·591	954	37·559
755	29·725	805	31·693	855	33·662	905	35·630	955	37·598
756	29·764	806	31·733	856	33·701	906	35·670	956	37·638
757	29·803	807	31·772	857	33·740	907	35·709	957	37·677
758	29·843	808	31·811	858	33·780	908	35·748	958	37·717
759	29·882	809	31·851	859	33·819	909	35·788	959	37·756
760	29·922	810	31·890	860	33·859	910	35·827	960	37·796
761	29·961	811	31·929	861	33·898	911	35·866	961	37·835
762	30·000	812	31·969	862	33·937	912	35·906	962	37·874
763	30·040	813	32·008	863	33·977	913	35·945	963	37·914
764	30·079	814	32·048	864	34·016	914	35·985	964	37·953
765	30·118	815	32·087	865	34·055	915	36·024	965	37·992
766	30·158	816	32·126	866	34·095	916	36·063	966	38·032
767	30·197	817	32·166	867	34·134	917	36·103	967	38·071
768	30·236	818	32·205	868	34·174	918	36·142	968	38·111
769	30·276	819	32·244	869	34·213	919	36·181	969	38·150
770	30·315	820	32·284	870	34·252	920	36·221	970	38·189
771	30·355	821	32·323	871	34·292	921	36·260	971	38·229
772	30·394	822	32·362	872	34·331	922	36·300	972	38·268
773	30·433	823	32·402	873	34·370	923	36·339	973	38·307
774	30·473	824	32·441	874	34·410	924	36·378	974	38·347
775	30·512	825	32·481	875	34·449	925	36·418	975	38·386
776	30·551	826	32·520	876	34·488	926	36·457	976	38·426
777	30·591	827	32·559	877	34·528	927	36·496	977	38·465
778	30·630	828	32·599	878	34·567	928	36·536	978	38·504
779	30·670	829	32·638	879	34·607	929	36·575	979	38·544
780	30·709	830	32·677	880	34·646	930	36·615	980	38·583
781	30·748	831	32·717	881	34·685	931	36·654	981	38·622
782	30·788	832	32·756	882	34·725	932	36·693	982	38·662
783	30·827	833	32·796	883	34·764	933	36·733	983	38·701
784	30·866	834	32·835	884	34·803	934	36·772	984	38·741
785	30·906	835	32·874	885	34·843	935	36·811	985	38·780
786	30·945	836	32·914	886	34·882	936	36·851	986	38·819
787	30·985	837	32·953	887	34·922	937	36·890	987	38·859
788	31·024	838	32·992	888	34·961	938	36·929	988	38·898
789	31·063	839	33·032	889	35·000	939	36·969	989	38·937
790	31·103	840	33·071	890	35·040	940	37·008	990	38·977
791	31·142	841	33·111	891	35·079	941	37·048	991	39·016
792	31·181	842	33·150	892	35·118	942	37·087	992	39·055
793	31·221	843	33·189	893	35·158	943	37·126	993	39·095
794	31·260	844	33·229	894	35·197	944	37·166	994	39·134
795	31·299	845	33·268	895	35·237	945	37·205	995	39·174
796	31·339	846	33·307	896	35·276	946	37·244	996	39·213
797	31·378	847	33·347	897	35·315	947	37·284	997	39·252
798	31·418	848	33·386	898	35·355	948	37·323	998	39·292
799	31·457	849	33·425	899	35·394	949	37·363	999	39·331
800	31·496	850	33·465	900	35·433	950	37·402	1000	39·370

Table 16.

Inches to Centimetres.

In.	0	1	2	3	4	5	6	7	8	9
	Cm.	Cm.	Cm.	Cm.	Cm.	Cm.	Cm.	Cm.	Cm.	Cm.
0	·0	2·54	5·08	7·62	10·16	12·7	15·24	17·78	20·32	22·86
10	25·4	27·94	30·48	33·02	35·56	38·1	40·64	43·18	45·72	48·26
20	50·8	53·34	55·88	58·42	60·96	63·5	66·04	68·58	71·12	73·66
30	76·2	78·74	81·28	83·82	86·36	88·9	91·44	93·98	96·52	99·06
40	101·6	104·14	106·68	109·22	111·76	114·3	116·84	119·38	121·92	124·46
50	127·0	129·54	132·08	134·62	137·16	139·7	142·24	144·78	147·32	149·86
60	152·4	154·94	157·48	160·02	162·56	165·1	167·64	170·18	172·72	175·26
70	177·8	180·34	182·88	185·42	187·96	190·5	193·04	195·58	198·12	200·66
80	203·2	205·74	208·28	210·82	213·36	215·9	218·44	220·98	223·52	226·06
90	228·6	231·14	233·68	236·22	238·76	241·3	243·84	246·38	248·92	251·46
100	254·0	256·54	259·08	261·62	264·16	266·7	269·24	271·78	274·32	276·86

4 inches = rather more than 10 centimetres

Table 17.

Centimetres to Inches.

Cm.	0	1	2	3	4	5	6	7	8	9
Cm.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
0	·000	·394	·787	1·181	1·575	1·969	2·362	2·756	3·150	3·543
10	3·937	4·331	4·742	5·118	5·512	5·906	6·299	6·693	7·087	7·480
20	7·874	8·268	8·662	9·055	9·449	9·843	10·236	10·630	11·024	11·418
30	11·811	12·205	12·599	12·992	13·386	13·780	14·173	14·567	14·961	15·355
40	15·748	16·142	16·536	16·929	17·323	17·717	18·111	18·504	18·898	19·292
50	19·685	20·079	20·473	20·867	21·260	21·654	22·048	22·441	22·835	23·229
60	23·622	24·016	24·410	24·804	25·197	25·591	25·985	26·378	26·772	27·166
70	27·560	27·953	28·347	28·741	29·134	29·528	29·922	30·316	30·709	31·103
80	31·497	31·890	32·284	32·678	33·071	33·465	33·859	34·253	34·646	35·040
90	35·434	35·827	36·221	36·615	37·009	37·402	37·796	38·190	38·583	38·977
100	39·370	39·764	40·158	40·552	40·945	41·339	41·733	42·126	42·520	42·914

2·5 centimetres or 25 millimetres = nearly 1 inch

TABLE 18.

EQUIVALENTS OF FEET, SQ. FEET, CU. FEET, IN METRES, SQ. METRES, AND CU. METRES.

Feet. Sq. Ft. Cu. Ft.	Metre.	Sq. Metre.	Cu. Metre.	Inches. Sq.Ins. Cu.Ins.	Cms.	Sq. Cms.	Cu. Cms.
1	0·3048	0·0929	0·0283	**1**	2·54	6·45	16·39
2	0·6096	0·1858	0·0566	**2**	5·08	12·90	32·77
3	0·9144	0·2787	0·0849	**3**	7·62	19·35	49·16
4	1·2192	0·3716	0·1133	**4**	10·16	25·81	65·55
5	1·5240	0·4645	0·1416	**5**	12·70	32·26	81·93
6	1·8288	0·5574	0·1699	**6**	15·24	38·71	98·32
7	2·1336	0·6503	0·1982	**7**	17·78	45·16	114·71
8	2·4384	0·7432	0·2265	**8**	20·32	51·61	131·09
9	2·7432	0·8361	0·2548	**9**	22·86	58·06	147·48
10	3·0479	0·9290	0·2832	**10**	25·40	64·51	163·86

TABLE 19.

EQUIVALENTS OF METRE, SQ. METRE, CU. METRE, IN FEET AND INCHES; SQ. FEET AND SQ. INCHES; CU. FEET AND CU. INCHES.

Metre. Sq. Met. Cu. Met.	Feet.	Inches.	Square Feet.	Square Inches.	Cubic Feet.	Cubic Inches.
1	3·281	39·37	10·76	1,550	35·32	61,026
2	6·562	78·74	21·53	3,100	70·63	122,052
3	9·843	118·11	32·29	4,650	105·95	183,078
4	13·124	157·48	43·06	6,200	141·27	244,103
5	16·404	196·85	53·82	7,750	176·58	305,129
6	19·685	236·23	64·59	9,300	211·90	366,155
7	22·966	275·60	75·36	10,850	247·22	427,183
8	26·247	314·97	86·11	12,400	282·53	488,201
9	29·528	354·34	96·88	13,950	317·85	549,237

ALGEBRA

Algebraic Expressions, Equations, Theorems and Series.

$+$	Plus, addition, positive, compression.		
$-$	Minus, subtraction, negative, tension.		
$=$	Equal to.	$\neq$	Not equal to.
$>$	Greater than.	$<$	Less than.
$\times$	Multiplied by.	$\div$	Divided by.
$\propto$	Proportional to.	∞	Infinity.
$\parallel$	Parallel to.	$\perp$	Perpendicular to.
$\because$	Because.	$\therefore$	Therefore.
$\angle$	Angle.	∟	Right Angle.
$\triangle$	Triangle.	$\odot$	Circle.
□″	Square inch.	m^2	Square metre.

a, b, c, and similar letters } at the beginning of alphabet denote known quantities.

$a^1, b^1, c^1,$ or $a'', b'', c'',$ or $a_1, b_1, c_1,$ } for corresponding quantities in like expressions.

x, y, z, and other letters } at the end of alphabet denote unknown quantities.

Other standard uses of symbols are—

c for a Constant.

d Differential and ∫ Integration in the Calculus.

F or *f* Function.

α, θ, φ Angles.

δ Variation.

Δ Finite Difference.

g Acceleration due to gravity (32·1908 ft. per sec. per sec. in London).

k a Coefficient.

Σ Sum of such quantities as

ϵ = 2·71828 base of Naperian or Hyperbolic logarithms.

Other standard uses of symbols—*contd.*

$\underline{|n}$ or $n!$, *i.e.* — Factorial n = product of first n numbers.

Example. $\underline{|4} = 1 \times 2 \times 3 \times 4 = 24$.

$\sqrt{\ }$ Square Root.

n^2 is n squared.

$\sqrt[3]{\ }$ cube root.

n^3 is n cubed.

π Ratio of circumference to diameter of a circle $= 3{\cdot}14159$ or $\frac{22}{7}$.

$(\ )$, $[\ ]$, $\{\ \}$. Brackets denoting that all terms inside must be dealt with together. For logarithm the abbreviation log is generally employed.

ALGEBRAIC RELATIONS.

$$\begin{aligned}
a^2 - b^2 &= (a + b)(a - b) \\
(a + b)^2 &= a^2 + 2ab + b^2 \\
(a - b)^2 &= a^2 - 2ab + b^2 \\
a^3 + b^3 &= (a + b)(a^2 - ab + b^2) \\
a^3 - b^3 &= (a - b)(a^2 + ab + b^2) \\
(a + b)^3 &= a^3 + 3a^2b + 3ab^2 + b^3 \\
(a - b)^3 &= a^3 - 3a^2b + 3ab^2 - b^3 \\
(x + a)(x + b) &= x^2 + (a + b)x + ab \\
(x + a)(x - b) &= x^2 + (a - b)x - ab
\end{aligned}$$

Proportion.

If a is to b as c is to d, which may be expressed

$$a : b :: c : d, \text{ then } \frac{a}{b} = \frac{c}{d}$$

and $a \times d = c \times b$, which latter may be expressed—

$$a{\cdot}d = c{\cdot}b \text{ or } ad = cb$$

Equivalent values for powers and roots—

$$a^m \times a^n = a^{m+n} \qquad a^m \div a^n = a^{m-n}$$

$$a^m \times b^m = (a{\cdot}b)^m \qquad a^m \div b^m = \left(\frac{a}{b}\right)^m$$

$$a^{-m} = \frac{1}{a^m} \qquad a^{\frac{1}{m}} = \sqrt[m]{a}$$

$$\sqrt[m]{a} \times \sqrt[m]{b} = \sqrt[m]{a{\cdot}b} \qquad \sqrt[m]{a} \div \sqrt[m]{b} = \sqrt[m]{\frac{a}{b}}$$

Permutations.

If any number of things be arranged in a number of ways each arrangement is called a Permutation.

The number of permutations of n things taken r at a time ${}^nPr$ is $n\ (n-1)\ (n-2)\ \ldots\ (n-r+2)\ (n-r+1)$.

If of the n things p are alike, q are alike, r are alike, then the permutations taken at a time are $\dfrac{\underline{|n}}{\underline{|p}\ \underline{|q}\ \underline{|r}}$

Combinations.

The combinations of a number of things are the different collections that can be formed without regard to the arrangement in each collection.

The combinations of n things taken r at a time ${}^nC_r = \dfrac{\underline{|n}}{\underline{|r}}$

EQUATIONS.

Simple.

$$ax + b = c$$

$$x = \frac{c-b}{a} \quad \ldots \quad (1)$$

$$x^n = d$$

$$x = \sqrt[n]{d} \quad \ldots \quad (2)$$

$$a^x = b$$

$$x \log a = \log b$$

$$x = \frac{\log b}{\log a} \quad \ldots \quad (3)$$

Simultaneous.

$$px + qy = r$$
$$sx + ty = v$$

$$x = \frac{tr - qv}{pt - qs} \text{ and } y = \frac{rs - pv}{qs - pt} \quad \ldots \quad (4)$$

Quadratic.

$$ax^2 + bx + c = 0$$

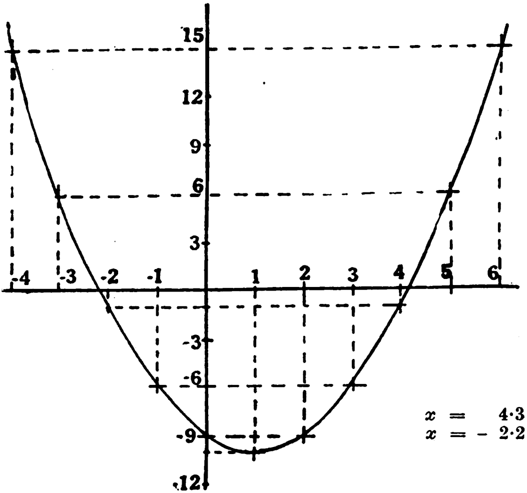
$$x = \frac{-b + \sqrt{b^2 - 4ac}}{2a} \text{ and } x = \frac{-b - \sqrt{b^2 - 4ac}}{2a} \quad \ldots \quad (5)$$

To solve an Equation using Graphs.

To find the values of x which makes $x^2 - 2x - 9 = 0$.

Make a table of values of $x^2 - 2x - 9$ for corresponding values of x.

$x =$	6	5	4	3	2	1	0	− 1	− 2	− 3	− 4
$x^2 - 2x - 9 =$	15	6	− 1	− 6	− 9	− 10	− 9	− 6	− 1	+ 6	15



Plot the values for $x^2 - 2 - 9$ above or below the plotted positions for x according as they are positive or negative, and draw a fair curve through as shown. The curve cuts the horizontal axis, whence $x^2 - 2x - 9 = 0$ at $+ 4\cdot 3$ and $- 2\cdot 2$. These are, therefore, the values of x which satisfy the equation.

Binomial Theorem.

The expansion of $(a + x)^n$ is—

$$(a + x)^n = a^n + n\, a^{n-1}x + \frac{n\,(n-1)}{1\cdot 2}\, a^{n-2}\, x^2 + \ldots$$

the rth term being

$$\frac{n\,(n-1)\,(n-2)\,\ldots\,(n-r+2)}{1\ \ 2\ \ 3\ \ \ldots\ \ (r-1)}\, a^{n-r+1}\, x^{r-1}$$

Exponential Theorem.

$$\text{The expansion of } a^x = 1 + x \log_\varepsilon a + \frac{(x \log_\varepsilon a)^2}{\underline{|2}} + \frac{(x \log_\varepsilon a)^3}{\underline{|2}} + \ldots$$

$$e^x = 1 + x + \frac{x^2}{\underline{|2}} + \frac{x^3}{\underline{|2}} + \ldots\ldots$$

$$e = 1 + 1 + \frac{1}{\underline{|2}} + \frac{1}{\underline{|3}} + \ldots\ldots$$

Logarithmic Series.

$$\mathrm{Log}_{\varepsilon}\,(1 + x) = x - \frac{x^2}{\underline{|2}} + \frac{x^3}{\underline{|3}} - \frac{x^4}{\underline{|4}} + \frac{x^5}{\underline{|5}} \ldots\ldots$$

Series in Arithmetical Progression.

$$a + (a + d) + (a + 2d) + (a + 3d) \ldots a + (n - 1)d$$

$$= na + d\left(\frac{n^2 - n}{2}\right)$$

Series in Geometrical Progression.

$$a + a{\cdot}d + a{\cdot}d^2 + ad^3 + \;..\; ad^{n-1} = \frac{a(d^n - 1)}{(d-1)}$$

General Series.

Sum to n Terms.

$$\Sigma n = 1 + 2 + 3 + 4 \ldots n = \frac{n^2}{2} + \frac{n}{2} = \frac{n(n+1)}{2}$$

$$\Sigma n^2 = 1^2 + 2^2 + 3^2 + 4^2 \ldots n^2 = \frac{n^3}{3} + \frac{n^2}{2} + \frac{n}{6} \text{ or}$$

$$\frac{n}{6}(n + 1)(2n + 1).$$

$$\Sigma n^3 = 1^3 + 2^3 + 3^3 + 4^3 \ldots n^3 = \frac{n^4}{4} + \frac{n^3}{2} + \frac{n^2}{4} \text{ or}$$

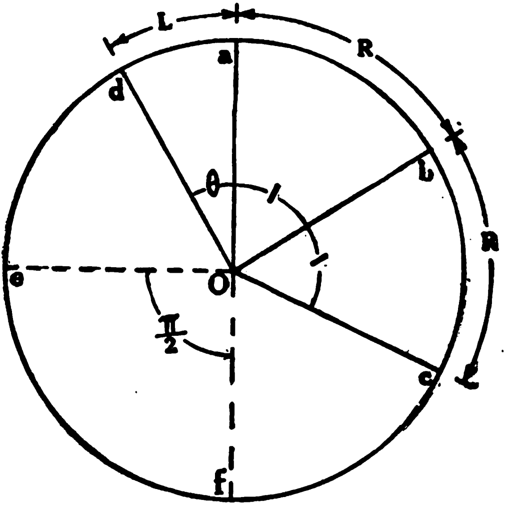
$$\frac{n^2}{4}(n^2 + 2n + 1) = (\Sigma n)^2.$$

TRIGONOMETRY.

Trigonometrical Functions and Relations. Plane Triangles.

The unit in which an angle can be expressed is not confined to the degree or to the right angle.

The "Radian" is another unit. It is the angle, such as *aOb*, subtended at the centre of a circle by an arc equal to the radius.



The angle *aOc* is 2 Radians.

$$\text{The angle } dOa = \frac{L}{R} \text{ Radians} = \theta$$

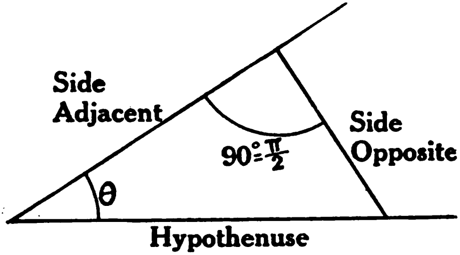
$$\text{The angle } fOe = 1 \text{ right angle} = \frac{\frac{\pi R}{2}}{R} = \frac{\pi}{2} \text{ Radians}$$

$$\text{or } 1 \text{ Radian} = \frac{2}{\pi} \text{ right angles}$$

$$= \frac{2 \times 90}{\pi} = 57{\cdot}2958 \text{ degrees}$$

Trigonometric Functions.

Using the two sides containing an angle θ, construct a right-angled triangle. The ratio of the three sides are constant for a given angle θ, and can therefore be used as functions of the angle.



Function.	Expressed.	Ratio.	Function.	Expressed.	Ratio.	Equivalent value
Sine	$\sin\theta$	$\frac{\text{side opposite}}{\text{Hypot.}}$	Cosecant	$\text{cosec}\,\theta$	$\frac{\text{Hypot.}}{\text{side opposite}}$	$\frac{1}{\sin\theta}$
Cosine	$\cos\theta$	$\frac{\text{side adjacent}}{\text{Hypot.}}$	Secant	$\sec\theta$	$\frac{\text{Hypot.}}{\text{side adjacent}}$	$\frac{1}{\cos\theta}$
Tangent	$\tan\theta$	$\frac{\text{side opposite}}{\text{side adjacent}}$	Cotangent	$\text{cotan}\,\theta$	$\frac{\text{side adjacent}}{\text{side opposite}}$	$\frac{1}{\tan\theta}$

General Trigonometric Relations.

The complement of an angle $\theta = 90^\circ - \theta$
„ supplement „ $\theta = 180^\circ - \theta$

$\sin^2\theta + \cos^2\theta = 1$; $\sin\theta \times \operatorname{cosec}\theta = 1$
$\tan\theta \times \operatorname{cotan}\theta = 1$; $\sec\theta \times \cos\theta = 1$

$\frac{\sin\theta}{\cos\theta} = \tan\theta$; $1 + \tan^2\theta = \sec^2\theta$

$\sqrt{1 - \sin^2\theta} = \cos\theta$; $\sqrt{1 - \cos^2\theta} = \sin\theta$
$\sec^2\theta = 1 + \tan^2\theta$; $\operatorname{cosec}^2\theta = 1 + \cot^2\theta$

$\sin\theta = \frac{\tan\theta}{\sqrt{1 + \tan^2\theta}}$; $\cos\theta = \frac{1}{\sqrt{1 + \tan^2\theta}}$

$\sin 2\theta = 2\sin\theta\cos\theta$; $\cos 2\theta = \cos^2\theta - \sin^2\theta$
$\cos 2\theta = 2\cos^2\theta - 1$; $\cos 2\theta = 1 - 2\sin^2\theta$

$\tan 2\theta = \frac{2\tan\theta}{1 - \tan^2\theta}$; $\cot 2\theta = \frac{\cot^2\theta - 1}{2\cot\theta}$

$\sin(\theta + \varphi) = \sin\theta\cos\varphi + \cos\theta\sin\varphi$
$\sin(\theta - \varphi) = \sin\theta\cos\varphi - \cos\theta\sin\varphi$
$\cos(\theta + \varphi) = \cos\theta\cos\varphi - \sin\theta\sin\varphi$
$\cos(\theta - \varphi) = \cos\theta\cos\varphi + \sin\theta\sin\varphi$

$2\sin\theta\cos\varphi = \sin(\theta + \varphi) + \sin(\theta - \varphi)$
$2\cos\theta\sin\varphi = \sin(\theta + \varphi) - \sin(\theta - \varphi)$
$2\sin\theta\sin\varphi = \cos(\theta - \varphi) - \cos(\theta + \varphi)$
$2\cos\theta\cos\varphi = \cos(\theta - \varphi) + \cos(\theta + \varphi)$

$\sin\theta + \sin\varphi = 2\sin\frac{\theta + \varphi}{2}\cos\frac{\theta - \varphi}{2}$

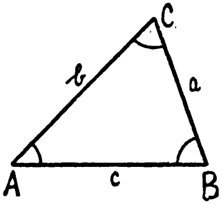
$\sin\theta - \sin\varphi = 2\cos\frac{\theta + \varphi}{2}\sin\frac{\theta - \varphi}{2}$

$\cos\theta + \cos\varphi = 2\cos\frac{\theta + \varphi}{2}\cos\frac{\theta - \varphi}{2}$

$\cos\theta - \cos\varphi = 2\sin\frac{\theta + \varphi}{2}\sin\frac{\theta - \varphi}{2}$

$\sin\theta = \cos(90 - \theta)$; $\cos\theta = \sin(90 - \theta)$;
$\tan\theta = \operatorname{cotan}(90 - \theta)$
$\sin(180 - \theta) = \sin\theta$; $\cos(180 - \theta) = -\cos\theta$;
$\tan(180 - \theta) = -\tan\theta$.
$\operatorname{cosec}(180 - \theta) = \operatorname{cosec}\theta$; $\sec(180 - \theta) = -\sec\theta$;
$\operatorname{cotan}(180 - \theta) = -\operatorname{cotan}\theta$.

Properties of Plane Triangles.



The area is denoted by the symbol S.
The half sum of the sides by s

$$= \frac{a + b + c}{2}$$

$$S = \sqrt{s\,(s - a)\,(s - b)\,(s - c)}$$

$$S = \frac{bc \sin A}{2} = \frac{ab \sin C}{2}$$

$$= \frac{ac \sin B}{2}$$

$$\frac{\sin A}{a} = \frac{\sin B}{b} = \frac{\sin C}{c}$$

$$\cos A = \frac{c^2 + b^2 - a^2}{2 \cdot cb}; \quad \cos B = \frac{a^2 + c^2 - b^2}{2ac};$$

$$\cos C = \frac{a^2 + b^2 - c^2}{2a.b}$$

$$\tan A = \frac{a \sin B}{c - a \cos B} = \frac{a \sin C}{b - a \cos c}$$

$$\sin \frac{A}{2} = \sqrt{\frac{(s - b)\,(s - c)}{bc}}; \quad \cos \frac{A}{2} = \sqrt{\frac{s\,(s - a)}{bc}};$$

$$\tan \frac{A}{2} = \sqrt{\frac{(s - b)\,(s - c)}{s\,(s - a)}}$$

$$\sin \frac{B}{2} = \sqrt{\frac{(s - a)\,(s - c)}{ac}}; \quad \cos \frac{B}{2} = \sqrt{\frac{s\,(s - b)}{ac}};$$

$$\tan \frac{B}{2} = \sqrt{\frac{(s - a)\,(s - c)}{s\,(s - b)}}$$

$$\sin \frac{C}{2} = \sqrt{\frac{(s - a)\,(s - b)}{ab}}; \quad \cos \frac{C}{2} = \sqrt{\frac{s\,(s - c)}{ab}};$$

$$\tan \frac{C}{2} = \sqrt{\frac{(s - a)\,(s - b)}{s\,(s - c)}}$$

Example. The sides of a triangle are 10″, 12″, 14″. Find the area.

$$s = \frac{10 + 12 + 14}{2} = 18$$

And the area $S = \sqrt{s\,(s - a)\,(s - b)\,(s - c)}$

$$= \sqrt{18 \times 8 \times 6 \times 4} = \sqrt{3456}$$

$$= 58 \cdot 8 \text{ sq. in.}$$

TABLE 20.
TRIGONOMETRICAL FUNCTIONS.

Angle.		Chord.	Sine.	Tan-gent.	Co-tan-gent.	Cosine.			
Deg-rees.	Ra-dians.								
0°	0	0	0	0	—	1	1·414	1·5708	90°
1	·0175	·017	·0175	·0175	57·2900	·9998	1·402	1·5533	89
2	·0349	·035	·0349	·0349	28·6363	·9994	1·389	1·5359	88
3	·0524	·052	·0523	·0524	19·0811	·9986	1·377	1·5184	87
4	·0698	·070	·0698	·0699	14·3007	·9976	1·364	1·5010	86
5	·0873	·087	·0872	·0875	11·4301	·9962	1·351	1·4835	85
6	·1047	·105	·1045	·1051	9·5144	·9945	1·338	1·4661	84
7	·1222	·122	·1219	·1228	8·1443	·9925	1·325	1·4486	83
8	·1396	·140	·1392	·1405	7·1154	·9903	1·312	1·4312	82
9	·1571	·157	·1564	·1584	6·3138	·9877	1·299	1·4137	81
10	·1745	·174	·1736	·1763	5·6713	·9848	1·286	1·3963	80
11	·1920	·192	·1908	·1944	5·1446	·9816	1·272	1·3788	79
12	·2094	·209	·2079	·2126	4·7046	·9781	1·259	1·3614	78
13	·2269	·226	·2250	·2309	4·3315	·9744	1·245	1·3439	77
14	·2443	·244	·2419	·2493	4·0108	·9703	1·231	1·3265	76
15	·2618	·261	·2588	·2679	3·7321	·9659	1·218	1·3090	75
16	·2793	·278	·2756	·2867	3·4874	·9613	1·204	1·2915	74
17	·2967	·296	·2924	·3057	3·2709	·9563	1·190	1·2741	73
18	·3142	·313	·3090	·3249	3·0777	·9511	1·176	1·2566	72
19	·3316	·330	·3256	·3443	2·9042	·9455	1·161	1·2392	71
20	·3491	·347	·3420	·3640	2·7475	·9397	1·147	1·2217	70
21	·3665	·364	·3584	·3831	2·6951	·9336	1·133	1·2043	69
22	·3840	·382	·3746	·4040	2·4751	·9272	1·118	1·1868	68
23	·4014	·399	·3907	·4245	2·3559	·9205	1·104	1·1694	67
24	·4189	·416	·4067	·4452	2·2460	·9135	1·089	1·1519	66
25	·4363	·433	·4226	·4663	2·1445	·9063	1·075	1·1345	65
26	·4538	·450	·4384	·4877	2·0503	·8988	1·060	1·1170	64
27	·4712	·467	·4540	·5095	1·9626	·8910	1·045	1·0996	63
28	·4887	·484	·4695	·5317	1·8807	·8829	1·030	1·0821	62
29	·5061	·501	·4848	·5543	1·8040	·8746	1·015	1·0647	61
30	·5236	·518	·5000	·5774	1·7321	·8660	1·000	1·0472	60
31	·5411	·534	·5150	·6009	1·6643	·8572	·985	1·0297	59
32	·5585	·551	·5299	·6249	1·6003	·8480	·970	1·0123	58
33	·5760	·568	·5446	·6494	1·5399	·8387	·954	·9948	57
34	·5934	·585	·5592	·6745	1·4826	·8290	·939	·9774	56
35	·6109	·601	·5736	·7002	1·4281	·8192	·923	·9599	55
36	·6283	·618	·5878	·7265	1·3764	·8090	·908	·9425	54
37	·6458	·635	·6018	·7536	1·3270	·7986	·892	·9250	53
38	·6632	·651	·6157	·7813	1·2799	·7880	·877	·9076	52
39	·6807	·668	·6293	·8098	1·2349	·7771	·861	·8901	51
40	·6981	·684	·6428	·8391	1·1918	·7660	·845	·8727	50
41	·7156	·700	·6561	·8693	1·1504	·7547	·829	·8552	49
42	·7330	·717	·6691	·9004	1·1106	·7431	·813	·8378	48
43	·7505	·733	·6820	·9325	1·0724	·7314	·797	·8203	47
44	·7679	·749	·6947	·9657	1·0355	·7193	·781	·8029	46
45°	·7854	·765	·7071	1·0000	1·0000	·7071	·765	·7854	45
			Cosine	Co-tan-gent.	Tan-gent.	Sine.	Chord.	Ra-dians.	Deg-rees.
								Angle.	

TABLE 21.

MINUTES AND SECONDS EXPRESSED IN DECIMALS OF A DEGREE

MIN.	0″	10″	20″	30″	40″	50″
0	·0	·00278	·00556	·00833	·01111	·01389
1	·01667	·01944	·02222	·025	·02778	·03055
2	·03333	·03611	·03888	·04166	·04444	·04722
3	·05	·05278	·05555	·05833	·06111	·06388
4	·06666	·06944	·07222	·075	·07777	·08055
5	·08333	·08611	·08888	·09166	·09444	·09722
6	·1	·1028	·1055	·1083	·1111	·1139
7	·1167	·1194	·1222	·125	·1278	·1306
8	·1333	·1361	·1388	·1417	·1444	·1472
9	·15	·1528	·1556	·1583	·1611	·1639
10	·1667	·1694	·1722	·175	·1778	·1805
11	·1833	·1861	·1889	·1917	·1944	·1972
12	·2	·2028	·2056	·2083	·2111	·2139
13	·2167	·2194	·2222	·225	·2278	·2306
14	·2333	·2361	·2389	·2417	·2444	·2472
15	·25	·2527	·2555	·2583	·2611	·2639
16	·2667	·2694	·2722	·275	·2778	·2806
17	·2833	·2861	·2889	·2917	·2944	·2972
18	·3	·3028	·3056	·3083	·3111	·3139
19	·3167	·3194	·3222	·325	·3278	·3306
20	·3333	·3361	·3389	·3417	·3444	·3472
21	·35	·3527	·3555	·3583	·3611	·3639
22	·3667	·3694	·3822	·375	·3778	·3806
23	·3833	·3861	·3889	·3917	·3944	·3972
24	·4	·4028	·4056	·4083	·4111	·4139
25	·4167	·4194	·4222	·425	·4278	·4306
26	·4333	·4361	·4389	·4417	·4444	·4472
27	·45	·4527	·4555	·4583	·4611	·4639
28	·4667	·4694	·4822	·475	·4778	·4806
29	·4833	·4861	·4889	·4917	·4944	·4972

TABLE 21 (*Continued*).

MINUTES AND SECONDS EXPRESSED IN DECIMALS OF A DEGREE.

MIN.	0″	10″	20″	30″	40″	50″
30	·5	·5028	·5056	·5083	·5111	·5139
31	·5167	·5194	·5222	·525	·5278	·5306
32	·5333	·5361	·5389	·5417	·5444	·5472
33	·55	·5527	·5555	·5583	·5611	·5639
34	·5667	·5694	·5822	·575	·5778	·5806
35	·5833	·5861	·5889	·5917	·5944	·5972
36	·6	·6028	·6056	·6083	·6111	·6139
37	·6167	·6194	·6222	·625	·6278	·6306
38	·6333	·6361	·6389	·6417	·6444	·6472
39	·65	·6527	·6555	·6583	·6611	·6639
40	·6667	·6694	·6822	·675	·6778	·6806
41	·6833	·6861	·6889	·6917	·6944	·6972
42	·7	·7028	·7056	·7083	·7111	·7139
43	·7167	·7194	·7222	·725	·7278	·7306
44	·7333	·7361	·7389	·7417	·7444	·7472
45	·75	·7527	·7555	·7583	·7611	·7639
46	·7667	·7694	·7822	·775	·7778	·7806
47	·7833	·7861	·7889	·7917	·7944	·7972
48	·8	·8028	·8056	·8083	·8111	·8139
49	·8167	·8194	·8222	·825	·8278	·8306
50	·8333	·8361	·8389	·8417	·8444	·8472
51	·85	·8527	·8555	·8583	·8611	·8639
52	·8667	·8694	·8822	·875	·8778	·8806
53	·8833	·8861	·8889	·8917	·8944	·8972
54	·9	·9028	·9056	·9083	·9111	·9139
55	·9167	·9194	·9222	·925	·9278	·9306
56	·9333	·9361	·9389	·9417	·9444	·9472
57	·95	·9527	·9555	·9583	·9611	·9639
58	·9667	·9694	·9822	·975	·9778	·9806
59	·9833	·9861	·9889	·9917	·9944	·9972

TABLE 22.

TAPERS AND ANGLES.

Taper Per Foot.	Complete Angle included.		Angle with Centre Line.		Taper Per Inch.	Taper Per Inch from Centre Line.
	Deg.	Min.	Deg.	Min.		
⅛	0	36	0	18	·010416	·005203
3/16	0	54	0	27	·015625	·007812
¼	1	12	0	36	·020833	·010416
5/16	1	30	0	45	·026042	·013021
⅜	1	47	0	53	·031250	·015625
7/16	2	05	1	02	·036458	·018229
½	2	23	1	11	·041667	·020833
9/16	2	42	1	21	·046875	·023438
⅝	3	00	1	30	·052084	·026042
11/16	3	18	1	39	·057292	·028646
¾	3	25	1	47	·062500	·031250
13/16	3	52	1	56	·067708	·033854
⅞	4	12	2	06	·072917	·036456
15/16	4	28	2	14	·078125	·039063
1	4	45	2	23	·083330	·041667
1¼	5	58	2	59	·104666	·052084
1½	7	08	3	34	·125000	·062500
1¾	8	20	4	10	·145833	·072917
2	9	32	4	46	·166666	·083332
2½	11	54	5	57	·208333	·104166
3	14	16	7	08	·250000	·125000
3½	16	36	8	18	·291666	·145833
4	18	54	9	27	·333333	·166666
4½	21	14	10	37	·375000	·187500
5	23	32	11	46	·416666	·208333
6	28	06	14	03	·500000	·250000

Standard Tapers—

Brown & Sharpe, ½″ per ft.; Morse, ⅝″ per foot.

ANALYTICAL GEOMETRY.

Certain regular forms can be represented by equations and plotted to co-ordinates x and y with respect to rectangular axes OX, OY.

The equation to a *straight line* is of the general form

$$y = mx + c,$$

where m is the tangent of the angle of inclination to the OX axis and c is the intercept on the OY axis.

THE CIRCLE.

The equation to a *circle* with origin at 0 is

$$x^2 + y^2 = R^2$$

where R is the radius.

If the centre is at a point represented by ordinates X and Y the equation is $(x - X)^2 + (y - Y)^2 = R^2$.

The equation to the tangent at a point $x^1\ y^1$ on the circle $x^2 + y^2 = R^2$ is $xx^1 + yy^1 = R^2$.

THE ELLIPSE.

The equation to an *ellipse* with centre at 0 is

$$\frac{x^2}{A^2} + \frac{y^2}{B^2} = 1$$

where A is half the major axis and B is half the minor axis.

The equation to a tangent at $x^1\ y^1$ is

$$\frac{x\,x^1}{A^2} + \frac{y\,y^1}{B^2} = 1$$

The equation to the normal at $x^1\ y^1$ is

$$y - y^1 = \frac{A^2}{B^2}\,\frac{y^1}{x^1}\,(x - x^1)$$

Note the tangent makes equal angles with the lines joining the point $x^1\ y^1$ with the foci.

The eccentricity of an ellipse $e = \frac{\sqrt{A^2 - B^2}}{A}$.

THE PARABOLA.

The equation to the *Parabola* with vertex at O is

$$y^2 = mx,$$

where $m = 2a$, a being the value of y for a point whose $x =$ the focal distance and $x = \frac{a}{2} = \frac{m}{4}$.

The equation for the tangent at $x^1\ y^1$ is

$$y\,y^1 = m\,(x - x^1)$$

The tangent makes equal angles with the axis and with the line joining the point $x^1\ y^1$ to the focus.

THE HYPERBOLA.

The equation to a *Hyperbola* with origin at the centre is

$$\frac{x^2}{A^2} - \frac{y^2}{B^2} = 1$$

$\pm A$ is the value of x of the vertex of each half.

The equation to the tangent at $x^1\ y^1$ is

$$\frac{x\ x^1}{A^2} - \frac{y\ y^1}{B^2} = 1$$

The equation to the normal at $x^1\ y^1$ is

$$A^2\ y^1 x + B^2\ x^1 y = (A^2 + B^2)\ x^1 y^1$$

The equation to a *Logarithmic Spiral* is

$$y = \log_a x$$

$$\text{or } a^y = x$$

The equation to the *Cissoid* is

$$y^2 = \frac{x^3}{2a - x}$$

The equation to a *catenary* is

$$y = \frac{a}{2}\left(\varepsilon^{\frac{x}{a}} + \varepsilon^{\frac{x}{a}}\right)$$

The equation to a *cycloid* is

$$x = a \text{ vers}^{-1} \frac{y}{a} + \sqrt{2\ xy - y^2}$$

$$\text{alternative form } x = a\ (\theta + \sin\theta)$$
$$y = a\ (1 - \cos\theta)$$

The *Epicycloid*

$$x = (a + b) \sin\theta - b \sin \frac{a + b}{b} \cdot \theta$$

$$y = (a + b) \cos\theta - b \cos \frac{a + b}{b} \cdot \theta$$

where a, b are radii of fixed and moving circles.

The *Hypocycloid.*

$$x = (a - b) \sin \theta - b \sin \frac{a - b}{b} \cdot \theta$$

$$y = a - b) \cos \theta + b \cos \frac{a - b}{b} \cdot \theta$$

The *Trochoid.*

$$x = a\,\theta - b \sin \theta$$
$$y = a - b \cos \theta$$

Involute of Circle.

$$x = a\,\theta \cos \theta - a \sin \theta$$
$$y = a \cos \theta + a\,\theta \sin \theta$$

The equation for the *Folium of Descartes*

$$x^3 + y^3 = 3\,axy$$

Polar Co-ordinates. When the locus of a point is expressed by R its distance from a point of origin O and by an angle θ which the line joining the point to the origin makes with an OX line the co-ordinates are known as Polar Co-ordinates.

The equations then become—

Line . . .	$R \cos (\theta - \alpha) = p$ (where p is the length of the perpendicular from the origin to the line, and α its inclination)
Ellipse . . .	$\frac{R^2 \cos^2 \theta}{A^2} + \frac{R^2 \sin^2 \theta}{B^2} = 1$
Parabola . . .	$R = a \sec^2 \frac{\theta}{2}$
Hyperbola . .	$\frac{R^2 \cos^2 \theta}{A^2} - \frac{R^2 \sin^2 \theta}{B^2} = 1$
Spiral of Archimedes .	$R = a\,\theta.$
Logarithmic Spiral .	$R = a^{\theta}$ or $ae^{m\theta}$
Hyperbolic Spiral .	$R\,\theta = a.$
Cardioid . . .	$R = a\,(1 - \cos \theta).$
Lemniscate . .	$R^2 = a^2 \cos 2\theta.$
Cissoid . . .	$R = \frac{2a \sin^2\theta}{\cos \theta}$

DIFFERENTIAL AND INTEGRAL CALCULUS

In an expression or equation wherein one quantity depends for its value upon the value of another quantity, the former is said to be a function of the latter. A function of x is expressed $f(x)$, and a corresponding general equation is $y = f(x)$.

The Differential Calculus deals with the relation between a difference in the value of y expressed dy corresponding to an exceedingly small difference in x expressed dx.

If a curve be plotted for the expression $y = f(x)$, then the value of $\frac{dy}{dx}$ is the tangent of the angle which a tangent to the curve makes with the OX axis.

General rules for differentiating—

$$y = \text{const} \qquad \frac{dy}{dx} = 0$$

$$y = x \qquad \frac{dy}{dx} = 1$$

$$y = x^n \qquad \frac{dy}{dx} = n \cdot x^{n-1}$$

$$y = u + v \qquad \frac{dy}{dx} = \frac{du}{dx} + \frac{dv}{dx}$$ where u and v are functions of x

$$y = u\,v \qquad \frac{dy}{dx} = u\,\frac{dv}{dx} + v\,\frac{du}{dx}$$

$$y = \text{const}\;u \qquad \frac{dy}{dx} = \text{const}\;\frac{du}{dx}$$

$$y = \frac{u}{v} \qquad \frac{dy}{dx} = \frac{v\,\frac{du}{dx} - u\,\frac{dv}{dx}}{v^2}$$

$$y = u^n \qquad \frac{dy}{dx} = n \cdot u^{n-1}\frac{du}{dx}$$

Standard Cases.

$$y = a^u \qquad \frac{dy}{dx} = a^u \log_e a \; \frac{du}{dx}$$

$$y = \log_a u \qquad \frac{dy}{dx} = \frac{1}{u} \log_a e \; \frac{du}{dx}$$

$$y = \sin u \qquad \frac{dy}{dx} = \cos u \frac{du}{dx}$$

$$y = \cos u \qquad \frac{dy}{dx} = -\sin u \frac{du}{dx}$$

$$y = \tan u \qquad \frac{dy}{dx} = \sec^2 u \frac{dy}{dx}$$

$$y = \operatorname{cosec} u \qquad \frac{dy}{dx} = -\operatorname{cosec} u \cot u \frac{du}{dx}$$

$$y = \sec u \qquad \frac{dy}{dx} = \sec u \tan u \frac{du}{dx}$$

$$y = \cot u \qquad \frac{dy}{dx} = -\operatorname{cosec}^2 u \frac{du}{dx}$$

$$y = \operatorname{vers} u \qquad \frac{dy}{dx} = \sin u \, du \frac{dx}{dn}$$

$$y = \sin^{-1} u \qquad \frac{dy}{dx} = \frac{1}{\sqrt{1 - u^2}} \frac{du}{dx}$$

$$y = \cos^{-1} u \qquad \frac{dy}{dx} = \frac{-1}{\sqrt{1 - u^2}} \frac{du}{dx}$$

$$y = \tan^{-1} u \qquad \frac{dy}{dx} = \frac{1}{1 + u^2} \frac{du}{dx}$$

MAXIMUM AND MINIMUM VALUES.

For obtaining values of x which give maximum or minimum values of y, solve the equation $\frac{dy}{dx} = 0$.

To distinguish between maxima and minima, find $\frac{d^2y}{dx^2}$ (*i.e.* the differential coefficient of $\frac{dy}{dx}$) and substitute the value of x already found. If the resultant expression is positive, y is minimum for this value of x, if the expression is negative y is at a maximum.

The Integral Calculus deals with the summation of infinitely small quantities. This operation, which is the reverse of differentiation, is denoted by the symbol $\int$ and the integration of ydx when limited to upper and lower values of x of a and δ is expressed $\int_{b}^{a} ydx$

Standard Cases.

$$\int a\, x^{n}\, dx = \frac{a.x^{n+1}}{n+1}$$

$$\int e^{x}\, dx = e^{x}$$

$$\int a^{x}\, dx = \frac{a^{n}}{\log\ a}$$

$$\int \frac{dx}{x} = \log\ x$$

$$\int \cos\ x\ dx = \sin\ x$$

$$\int \sin\ x\ dx = -\cos\ x$$

$$\int \frac{dx}{x^{2}+a^{2}} = \frac{1}{a}\ \tan^{-1}\ \frac{a}{x}$$

$$\int \frac{dx}{x^{2}-a^{2}} = \frac{1}{2a}\ \log\ \frac{x-a}{x+a}$$

$$\int \frac{dx}{\sqrt{a^{2}-x^{2}}} = \sin^{-1}\ \frac{x}{a}$$

$$\int \frac{dx}{a\sqrt{x^{2}-a^{2}}} = \frac{1}{a}\ \sec^{-1}\ \frac{x}{a}$$

LOGARITHMS (Logs.).

$10000{\cdot}0000 = 10^{+4}$
$1000{\cdot}0000 = 10^{+3}$
$100{\cdot}0000 = 10^{+2}$
$10{\cdot}0000 = 10^{+1}$
$1{\cdot}0000 = 10^{0}$
${\cdot}1000 = 10^{-1}$
${\cdot}0100 = 10^{-2}$
${\cdot}0010 = 10^{-3}$

The common logarithm of a number is the index to which the base 10 is raised to give the number. For example, 10 to the power of 2 (10^2) that is 10 squared equals 100, so that 2 is the common logarithm of 100.

$2896{\cdot}0000 = 10^{3.4618}$
$289{\cdot}6000 = 10^{2.4618}$
$28{\cdot}9600 = 10^{1.4618}$
$2{\cdot}8960 = 10^{0.4618}$
${\cdot}2896 = 10^{\bar{1}.4618}$

The logarithmic tables supply only the decimal part of a logarithm, known as the *mantissa*, and this decimal part is always a + quantity.

For example, consider the number 2896. Referring to the tables 2896, we get a decimal value 4609 + 9 = ·4618; and the whole number portion or *characteristic* is obtained as indicated in the adjoining table. The negative (–) characteristic is indicated by a bar over the whole number.

Examples. The logarithm of 6065 is made up of a whole number 3, because it is between 1000 = 10^3 and 10000 = 10^4, and the decimal part is 7829 + 4 = ·7829.

Expressed formally, log 6065 = 3·7829.

Verify—log 89·33 = 1·9510; log ·4226 = 1·6259.

Rule 1. To multiply two or more numbers, the sum of the logs added together equals the log of the product.

Example. 89·33 × ·4226.

Log 89·33 = 1·9510
Log ·4226 = $\bar{1}{\cdot}6259$

Sum logs = 1·5769 = log of Product
Product = 37·75

RULE 2. To divide one number into another, the difference of the logs equals the log of the quotient.

Example. 89·33 ÷ ·4226

Log 89·33 = 1·9510

Log ·4226 = $\bar{1}$·6259

Diffce. Logs = 2·3251 = log of quotient

Quotient = 211·4

RULE 3. To find the square or Cube of a number. The product of the log thereof, when multiplied by 2 or 3 respectively, is the logarithm of the answer.

Example. $4{\cdot}226^3$

Log 4·226 = 0·6259

3

Log answer = 1·8777

Answer = 75·44

RULE 4. To find the square or cube root of a number. The quotient of the log thereof, when divided by 2 or 3 respectively, is the log of the answer.

Example. $\sqrt{893{\cdot}3}$

Log 893·3 = 2·9510

Dividing by 2 = Log answer = 1·4755

Answer = 29·89

RULE 4 (special case). When taking the root of a number and the characteristic is negative and not exactly divisible, the characteristic should be made up to be exactly divisible and the negative quantity so used must be added to the mantissa.

Example. $\sqrt{\cdot 8933}$

Log .8933 = $\bar{1}$·9510

= $\bar{2}$ + 1·9510

Dividing by 2 *B* Log answer = $\bar{1}$·9755

Answer = ·9451

Example. $\sqrt[5]{\cdot 008933}$

Log ·008933 = $\bar{3}$·9510

= $\bar{5}$ + 2·9510

Dividing by 5 = Log answer = $\bar{1}$·5902

Answer = ·3893

To convert Common Logarithms to Naperian or Hyperbolic, multiply by 2·3026.

This is useful when determining the Mean Effective Pressure in a steam cylinder, using only Common Logarithms.

When P is the Initial Pressure in a cylinder in lbs. per sq. in.,
p ,, Back ,, ,, ,, ,,
and r ,, ratio of expansion,

Then the Mean Effective Pressure $= \text{P}\left(\dfrac{1 + \log_e r}{r}\right) - p$

$$\text{or M.E.P.} = \text{P}\left(\frac{1 + 2{\cdot}3026 \times \log_{10} r}{r}\right) - p$$

Example. P = 150; p = 25 and r = 3

$$\text{Then M.E.P.} = 150\left(\frac{1 + 2{\cdot}3026 \log 3}{3}\right) - 25$$

$$= 150\left(\frac{1 + (2{\cdot}3026 \times {\cdot}4771)}{3}\right) - 25$$

$$= 150\left(\frac{1 + 1{\cdot}098}{3}\right) - 25$$

$$= \frac{150 \times 2{\cdot}098}{3} - 25$$

$$= 80 \text{ lbs. per sq. in.}$$

TABLE 23.

COMMON LOGARITHMS.

No.	0	1	2	3	4	5	6	7	8	9	Diff.
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	43—40
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	39—36
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	36—34
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	34—31
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	31—29
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	29—27
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	27—26
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	26—25
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	24—23
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	23—22
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	22—20
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	21—19
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	20—19
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	19—18
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	18—17
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	18—17
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	17—16
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	16—15
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	16—15
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	15—14
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	15—14
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	14—13
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	14—13
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	13
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	13—12
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	13—12
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	12—11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	12—11
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	12—11
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	12—11
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	11—10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	11—10
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	11—10
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	10
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	10—9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	10—9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	10—9
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	10—9
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	9
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	9—8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	9—8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	9—8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	9—8
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	9—8
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	8
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	8—7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	8—7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	8—7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	8—7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	8—7
No.	0	1	2	3	4	5	6	7	8	9	Diff.

TABLE 23.

COMMON LOGARITHMS (*Continued*).

No.	0	1	2	3	4	5	6	7	8	9	Diff.
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	8—7
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	8—7
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	7—6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	7—6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	7—6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	7—6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	7—6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	7—6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	7—6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	7—6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	7—6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	7—6
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	6
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	6—5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	6—5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	6—5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	6—5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	6—5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	6—5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	6—5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	6—5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	6—5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	6—5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	6—5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	6—5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	6—5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	5
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	5—4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	5—4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	5—4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	5—4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	5—4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	5—4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	5—4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	5—4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	5—4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	5—4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	5—4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	5—4
100	00000	00043	00087	00130	00173	00217	00260	00303	00346	00389	43
101	00432	00475	00518	00561	00604	00647	00689	00732	00775	00817	43
102	00860	00903	00945	00988	01030	01072	01115	01157	01199	01242	43
103	01284	01326	01368	01410	01452	01494	01536	01578	01620	01662	42
104	01703	01745	01787	01828	01870	01912	01953	01995	02036	02078	42
105	02119	02160	02202	02243	02284	02325	02366	02408	02449	02490	41
106	02531	02572	02612	02653	02694	02735	02776	02816	02857	02898	41
107	02938	02979	03019	03060	03100	03141	03181	03222	03262	03302	41—40
108	03342	03383	03423	03463	03503	03543	03583	03623	03663	03703	41—40
109	03743	03782	03822	03862	03902	03941	03981	04021	04060	04100	41—40
No.	0	1	2	3	4	5	6	7	8	9	Diff.

TABLE 24.

ANTILOGARITHMS.

Log.	0	1	2	3	4	5	6	7	8	9	Diff.
00	1000	1002	1005	1007	1009	1012	1014	1016	1019	1021	2—3
01	1023	1026	1028	1030	1033	1035	1038	1040	1042	1045	2—3
02	1047	1050	1052	1054	1057	1059	1062	1064	1067	1069	2—3
03	1072	1074	1076	1079	1081	1084	1086	1089	1091	1094	2—3
04	1096	1099	1102	1104	1107	1109	1112	1114	1117	1119	2—3
05	1122	1125	1127	1130	1132	1135	1138	1140	1143	1146	2—3
06	1148	1151	1153	1156	1159	1161	1164	1167	1169	1172	2—3
07	1175	1178	1180	1183	1186	1189	1191	1194	1197	1199	2—3
08	1202	1205	1208	1211	1213	1216	1219	1222	1225	1227	2—3
09	1230	1233	1236	1239	1242	1245	1247	1250	1253	1256	2—3
·10	1259	1262	1265	1268	1271	1274	1276	1279	1282	1285	2—3
11	1288	1291	1294	1297	1300	1303	1306	1309	1312	1315	3
12	1318	1321	1324	1327	1330	1334	1337	1340	1343	1346	3—4
13	1349	1352	1355	1358	1361	1365	1368	1371	1374	1377	3—4
14	1380	1384	1387	1390	1393	1396	1400	1403	1406	1409	3—4
15	1413	1416	1419	1422	1426	1429	1432	1435	1439	1442	3—4
16	1445	1449	1452	1455	1459	1462	1466	1469	1472	1476	3—4
17	1479	1483	1486	1489	1493	1496	1500	1503	1507	1510	3—4
18	1514	1517	1521	1524	1528	1531	1535	1538	1542	1545	3—4
19	1549	1552	1556	1560	1563	1567	1570	1574	1578	1581	3—4
20	1585	1589	1592	1596	1600	1603	1607	1611	1614	1618	3—4
21	1622	1626	1629	1633	1637	1641	1644	1648	1652	1656	3—4
22	1660	1663	1667	1671	1675	1679	1683	1687	1690	1694	3—4
23	1698	1702	1706	1710	1714	1718	1722	1726	1730	1734	3—4
24	1738	1742	1746	1750	1754	1758	1762	1766	1770	1774	4
25	1778	1782	1786	1791	1795	1799	1803	1807	1811	1816	4—5
26	1820	1824	1828	1832	1837	1841	1845	1849	1854	1858	4—5
27	1862	1866	1871	1875	1879	1884	1888	1892	1897	1901	4—5
28	1905	1910	1914	1919	1923	1928	1932	1936	1941	1945	4—5
29	1950	1954	1959	1963	1968	1972	1977	1982	1986	1991	4—5
30	1995	2000	2004	2009	2014	2018	2023	2028	2032	2037	4—5
31	2042	2046	2051	2056	2061	2065	2070	2075	2080	2084	4—5
32	2089	2094	2099	2104	2109	2113	2118	2123	2128	2133	4—5
33	2138	2143	2148	2153	2158	2163	2168	2173	2178	2183	4—5
34	2188	2193	2198	2203	2208	2213	2218	2223	2228	2234	5
35	2239	2244	2249	2254	2259	2265	2270	2275	2280	2286	5—6
36	2291	2296	2301	2307	2312	2317	2323	2328	2333	2339	5—6
37	2344	2350	2355	2360	2366	2371	2377	2382	2388	2393	5—6
38	2399	2404	2410	2415	2421	2427	2432	2438	2443	2449	5—6
39	2455	2460	2466	2472	2477	2483	2489	2495	2500	2506	5—6
40	2512	2518	2523	2529	2535	2541	2547	2553	2559	2564	5—6
41	2570	2576	2582	2588	2594	2600	2606	2612	2618	2624	6
42	2630	2636	2642	2649	2655	2661	2667	2673	2679	2685	6—7
43	2692	2698	2704	2710	2716	2723	2729	2735	2742	2748	6—7
44	2754	2761	2767	2773	2780	2786	2793	2799	2805	2812	6—7
45	2818	2825	2831	2838	2844	2851	2858	2864	2871	2877	6—7
46	2884	2891	2897	2904	2911	2917	2924	2931	2938	2944	6—7
47	2951	2958	2965	2972	2979	2985	2992	2999	3006	3013	6—7
48	3020	3027	3034	3041	3048	3055	3062	3069	3076	3083	7
49	3090	3097	3105	3112	3119	3126	3133	3141	3148	3155	7—8
50	3162	3170	3177	3184	3192	3199	3206	3214	3221	3228	7—8
51	3236	3243	3251	3258	3266	3273	3281	3289	3296	3304	7—8
52	3311	3319	3327	3334	3342	3350	3357	3365	3373	3381	7—8
53	3388	3396	3404	3412	3420	3428	3436	3443	3451	3459	7—8
54	3467	3475	3483	3491	3499	3508	3516	3524	3532	3540	8—9
Log.	0	1	2	3	4	5	6	7	8	·9	Diff.

TABLE 24.

ANTILOGARITHMS (*Continued*).

Log.	0	1	2	3	4	5	6	7	8	9	Diff.
55	3548	3556	3565	3573	3581	3589	3597	3606	3614	3622	8—9
56	3631	3639	3648	3656	3664	3673	3681	3690	3698	3707	8—9
57	3715	3724	3733	3741	3750	3758	3767	3776	3784	3793	8—9
58	3802	3811	3819	3828	3837	3846	3855	3864	3873	3882	8—9
59	3890	3899	3908	3917	3926	3936	3945	3954	3963	3972	9—10
60	3981	3990	3999	4009	4018	4027	4036	4046	4055	4064	9—10
61	4074	4083	4093	4102	4111	4121	4130	4140	4150	4159	9—10
62	4169	4178	4188	4198	4207	4217	4227	4236	4246	4256	9—10
63	4266	4276	4285	4295	4305	4315	4325	4335	4345	4355	9—10
64	4365	4375	4385	4395	4406	4416	4426	4436	4446	4457	10—11
65	4467	4477	4487	4498	4508	4519	4529	4539	4550	4560	10—11
66	4571	4581	4592	4603	4613	4624	4634	4645	4656	4667	10—11
67	4677	4688	4699	4710	4721	4732	4742	4753	4764	4775	10—11
68	4786	4797	4808	4819	4831	4842	4853	4864	4875	4887	11—12
69	4898	4909	4920	4932	4943	4955	4966	4977	4989	5000	11—12
70	5012	5023	5035	5047	5058	5070	5082	5093	5105	5117	11—12
71	5129	5140	5152	5164	5176	5188	5200	5212	5224	5236	11—12
72	5248	5260	5272	5284	5297	5309	5321	5333	5346	5358	12—13
73	5370	5383	5395	5408	5420	5433	5445	5458	5470	5483	12—13
74	5495	5508	5521	5534	5546	5559	5572	5585	5598	5610	12—13
75	5623	5636	5649	5662	5675	5689	5702	5715	5728	5741	13—14
76	5754	5768	5781	5794	5808	5821	5834	5848	5861	5875	13—14
77	5888	5902	5916	5929	5943	5957	5970	5984	5998	6012	13—14
78	6026	6039	6053	6067	6081	6095	6109	6124	6138	6152	13—14
79	6166	6180	6194	6209	6223	6237	6252	6266	6281	6295	14—15
80	6310	6324	6339	6353	6368	6383	6397	6412	6427	6442	14—15
81	6457	6471	6486	6501	6516	6531	6546	6561	6577	6592	14—15
82	6607	6622	6637	6653	6668	6683	6699	6714	6730	6745	15—16
83	6761	6776	6792	6808	6823	6839	6855	6871	6887	6902	15—16
84	6918	6934	6950	6966	6982	6998	7015	7031	7047	7063	16—17
85	7079	7096	7112	7129	7145	7161	7178	7194	7211	7228	16—17
86	7244	7261	7278	7295	7311	7328	7345	7362	7379	7396	16—17
87	7413	7430	7447	7464	7482	7499	7516	7534	7551	7568	17—18
88	7586	7603	7621	7638	7656	7674	7691	7709	7727	7745	18—19
89	7762	7780	7798	7816	7834	7852	7870	7889	7907	7925	18—19
90	7943	7962	7980	7998	8017	8035	8054	8072	8091	8110	18—19
91	8128	8147	8166	8185	8204	8222	8241	8260	8279	8299	18—19
92	8318	8337	8356	8375	8395	8414	8433	8453	8472	8492	19—20
93	8511	8531	8551	8570	8590	8610	8630	8650	8670	8690	19—20
94	8710	8730	8750	8770	8790	8810	8831	8851	8872	8892	20—21
95	8913	8933	8954	8974	8995	9016	9036	9057	9078	9099	20—21
96	9120	9141	9162	9183	9204	9226	9247	9268	9290	9311	21—22
97	9333	9354	9376	9397	9419	9441	9462	9484	9506	9528	21—22
98	9550	9572	9594	9616	9638	9661	9683	9705	9727	9750	22—23
99	9772	9795	9817	9840	9863	9886	9908	9931	9954	9977	22—23
00	10000	10023	10046	10069	10093	10116	10139	10162	10186	10209	23—24
01	10233	10257	10280	10304	10328	10351	10375	10399	10423	10447	23—25
02	10471	10495	10520	10544	10568	10593	10617	10641	10666	10691	24—25
03	10715	10740	10765	10789	10814	10839	10864	10889	10914	10940	24—25
04	10965	10990	11015	11041	11066	11092	11117	11143	11169	11194	25—26
05	11220	11246	11272	11298	11324	11350	11376	11402	11429	11455	26—27
06	11482	11508	11535	11561	11588	11614	11641	11668	11695	11722	26—27
07	11749	11776	11803	11830	11858	11885	11912	11940	11967	11905	27—28
08	12023	12050	12078	12106	12134	12162	12190	12218	12246	12274	27—28
09	12303	12331	12359	12388	12417	12445	12474	12503	12531	12560	28—29
Log.	0	1	2	3	4	5	6	7	8	9	Diff.

TABLE 25.
LOGARITHMS OF TRIGONOMETRICAL RATIOS.

Deg.	Min.	L. Sin.	Diff.	L. Cos.	L. Tan.	Diff.	L. Cot.		
0	0	neg. inf.		0·0000	neg. inf.		pos. inf.	90	0
	10	7·4637	3011	0·0000	7·4637	3011	2·5363	89	50
	20	7·7648	1760	0·0000	7·7648	1761	2·2352		40
	30	7·9408	1250	0·0000	7·9409	1249	2·0591		30
	40	8·0658	969	0·0000	8·0658	969	1·9342		20
	50	8·1627	792	0·0000	8·1627	792	1·8373		10
1	0	8·2419	669	9·9999	8·2419	670	1·7581	89	0
	10	8·3088	580	9·9999	8·3089	580	1·6911	88	50
	20	8·3668	511	9·9999	8·3669	512	1·6331		40
	30	8·4179	458	9·9999	8·4181	457	1·5819		30
	40	8·4637	413	9·9998	8·4638	415	1·5362		20
	50	8·5050	378	9·9998	8·5053	378	1·4947		10
2	0	8·5428	348	9·9997	8·5431	348	1·4569	88	0
	10	8·5776	321	9·9997	8·5779	322	1·4221	87	50
	20	8·6097	300	9·9996	8·6101	300	1·3899		40
	30	8·6397	280	9·9996	8·6401	281	1·3599		30
	40	8·6677	263	9·9995	8·6682	263	1·3318		20
	50	8·6940	248	9·9995	8·6945	249	1·3055		10
3	0	8·7188	235	9·9994	8·7194	235	1·2806	87	0
	10	8·7423	222	9·9993	8·7429	223	1·2571	86	50
	20	8·7645	212	9·9993	8·7652	213	1·2348		40
	30	8·7857	202	9·9992	8·7865	202	1·2135		30
	40	8·8059	192	9·9991	8·8067	194	1·1933		20
	50	8·8251	185	9·9990	8·8261	185	1·1739		10
4	0	8·8436	177	9·9989	8·8446	178	1·1554	86	0
	10	8·8613	170	9·9989	8·8624	171	1·1376	85	50
	20	8·8783	163	9·9988	8·8795	165	1·1205		40
	30	8·8946	158	9·9987	8·8960	158	1·1040		30
	40	8·9104	152	9·9986	8·9118	154	1·0882		20
	50	8·9256	147	9·9985	8·9272	148	1·0728		10
5	0	8·9403	142	9·9983	8·9420	143	1·0580	85	0
	10	8·9545	137	9·9982	8·9563	138	1·0437	84	50
	20	8·9682	134	9·9981	8·9701	135	1·0299		40
	30	8·9816	129	9·9980	8·9836	130	1·0164		30
	40	8·9945	125	9·9979	8·9966	127	1·0034		20
	50	9·0070	122	9·9977	9·0093	123	0·9907		10
6	0	9·0192	119	9·9976	9·0216	120	0·9784	84	0
	10	9·0311	115	9·9975	9·0336	117	0·9664	83	50
	20	9·0426	113	9·9973	9·0453	114	0·9547		40
	30	9·0539	109	9·9972	9·0567	111	0·9433		30
	40	9·0648	107	9·9971	9·0678	108	0·9322		20
	50	9·0755	104	9·9969	9·0786	105	0·9214		10
7	0	9·0859	102	9·9968	9·0891	104	0·9109	83	0
	10	9·0961	99	9·9966	9·0995	101	0·9005	82	50
	20	9·1060		9·9964	9·1096		0·8904		40
		L. Cos.	Diff.	L. Sin.	L. Cot.	Diff.	L. Tan.	Deg.	Min.

TABLE 25.

LOGARITHMS OF TRIGONOMETRICAL RATIOS (*Continued*).

Deg.	Min.	L. Sin.	Diff.	L. Cos.	L. Tan.	Diff.	L. Cot.		
7	30	9·1157	95	9·9963	9·1194	97	0·8806	82	30
	40	9·1252	93	9·9961	9·1291	94	0·8709		20
	50	9·1345	91	9·9959	9·1385	93	0·8615		10
8	0	9·1436	89	9·9958	9·1478	91	0·8522	82	0
	10	9·1525	87	9·9956	9·1569	89	0·8431	81	50
	20	9·1612	85	9·9954	9·1658	87	0·8342		40
	30	9·1697	84	9·9952	9·1745	86	0·8255		30
	40	9·1781	82	9·9950	9·1831	84	0·8169		20
	50	9·1863	80	9·9948	9·1915	82	0·8085		10
9	0	9·1943	79	9·9946	9·1997	81	0·8003	81	0
	10	9·2022	78	9·9944	9·2078	80	0·7922	80	50
	20	9·2100	76	9·9942	9·2158	78	0·7842		40
	30	9·2176	75	9·9940	9·2236	77	0·7764		30
	40	9·2251	73	9·9938	9·2313	76	0·7687		20
	50	9·2324	73	9·9936	9·2389	74	0·7611		10
10	0	9·2397	71	9·9934	9·2463	73	0·7537	80	0
	10	9·2468	70	9·9931	9·2536	73	0·7464	79	50
	20	9·2538	68	9·9929	9·2609	71	0·7391		40
	30	9·2606	68	9·9927	9·2680	70	0·7320		30
	40	9·2674	66	9·9924	9·2750	69	0·7250		20
	50	9·2740	66	9·9922	9·2819	68	0·7181		10
11	0	9·2806	64	9·9919	9·2887	66	0·7113	79	0
	10	9·2870	64	9·9917	9·2953	67	0·7047	78	50
	20	9·2934	63	9·9914	9·3020	65	0·6980		40
	30	9·2997	61	9·9912	9·3085	64	0·6915		30
	40	9·3058	61	9·9909	9·3149	63	0·6851		20
	50	9·3119	60	9·9907	9·3212	63	0·6788		10
12	0	9·3179	59	9·9904	9·3275	61	0·6725	78	0
	10	9·3238	58	9·9901	9·3336	61	0·6664	77	50
	20	9·3296	57	9·9899	9·3397	61	0·6603		40
	30	9·3353	57	9·9896	9·3458	59	0·6542		30
	40	9·3410	56	9·9893	9·3517	59	0·6483		20
	50	9·3466	55	9·9890	9·3576	58	0·6424		10
13	0	9·3521	54	9·9887	9·3634	57	0·6366	77	0
	10	9·3575	54	9·9884	9·3691	57	0·6309	76	50
	20	9·3629	53	9·9881	9·3748	56	0·6252		40
	30	9·3682	52	9·9878	9·3804	55	0·6196		30
	40	9·3734	52	9·9875	9·3859	55	0·6141		20
	50	9·3786	51	9·9872	9·3914	54	0·6086		10
14	0	9·3837	50	9·9869	9·3968	53	0·6032	76	0
	10	9·3887	50	9·9866	9·4021	53	0·5979	75	50
	20	9·3937	49	9·9863	9·4074	53	0·5926		40
	30	9·3986	49	9·9859	9·4127	51	0·5873		30
	40	9·4035	48	9·9856	9·4178	52	0·5822		20
	50	9·4083		9·9853	9·4230		0·5770		10
		L. Cos.	Diff.	L. Sin.	L. Cot.	Diff.	L. Tan.	Deg.	Min.

TABLE 25.

LOGARITHMS OF TRIGONOMETRICAL RATIOS (*Continued*).

Deg.	Min.	L. Sin.	Diff.	L. Cos.	Diff.	L. Tan.	Diff.	L. Cot.		
15	0	9·4130		9·9849		9·4281		0·5719	75	0
	10	9·4177	47	9·9846	3	9·4331	50	0·5669	74	50
	20	9·4223	46	9·9843	3	9·4381	50	0·5619		40
	30	9·4269	46	9·9839	4	9·4430	49	0·5570		30
	40	9·4314	45	9·9836	3	9·4479	49	0·5521		20
	50	9·4359	45	9·9832	4	9·4527	48	0·5473		10
16	0	9·4403	44	9·9828	4	9·4575	48	0·5425	74	0
	10	9·4447	44	9·9825	3	9·4622	47	0·5378	73	50
	20	9·4491	44	9·9821	4	9·4669	47	0·5331		40
	30	9·4533	42	9 9817	4	9·4716	47	0·5284		30
	40	9·4576	43	9·9814	3	9·4762	46	0·5238		20
	50	9·4618	42	9·9810	4	9·4808	46	0·5192		10
17	0	9·4659	41	9·9806	4	9·4853	45	0·5147	73	0
	10	9·4700	41	9·9802	4	9·4898	45	0·5102	72	50
	20	9·4741	41	9·9798	4	9·4943	45	0 5057		40
	30	9·4781	40	9·9794	4	9·4987	44	0·5013		30
	40	9·4821	40	9·9790	4	9·5031	44	0·4969		20
	50	9·4861	40	9·9786	4	9·5075	44	0·4925		10
18	0	9·4900	39	9·9782	4	9·5118	43	0·4882	72	0
	10	9·4939	39	9·9778	4	9·5161	43	0·4839	71	50
	20	9·4977	38	9·9774	4	9·5203	42	0·4797		40
	30	9·5015	38	9·9770	4	9·5245	42	0·4755		30
	40	9·5052	37	9·9765	5	9·5287	42	0·4713		20
	50	9·5090	38	9·9761	4	9·5329	42	0·4671		10
19	0	9·5126	36	9·9757	4	9·5370	41	0·4630	71	0
	10	9·5163	37	9·9752	5	9·5411	41	0·4589	70	50
	20	9·5199	36	9·9748	4	9·5451	40	0·4549		40
	30	9·5235	36	9·9743	5	9·5491	40	0·4509		30
	40	9·5270	35	9·9739	4	9·5531	40	0·4469		20
	50	9·5306	36	9·9734	5	9·5571	40	0·4429		10
20	0	9·5341	35	9·9730	4	9·5611	40	0·4389	70	0
	10	9·5375	34	9·9725	5	9·5650	39	0·4350	69	50
	20	9·5409	34	9·9721	4	9·5689	39	0·4311		40
	30	9·5443	34	9·9716	5	9·5727	38	0·4273		30
	40	9·5477	34	9·9711	5	9·5766	39	0·4234		20
	50	9·5510	33	9·9706	5	9·5804	38	0·4196		10
21	0	9·5543	33	9·9702	4	9·5842	38	0·4158	69	0
	10	9·5576	33	9·9697	5	9·5879	37	0·4121	68	50
	20	9·5609	33	9·9692	5	9·5917	38	0·4083		40
	30	9·5641	32	9·9687	5	9·5954	37	0·4046		30
	40	9·5673	32	9·9682	5	9·5991	37	0·4009		20
	50	9·5704	31	9·9677	5	9·6028	37	0·3972		10
22	0	9·5736	32	9·9672	5	9·6064	36	0·3936	68	0
	10	9·5767	31	9·9667	5	9·6100	36	0·3900	67	50
	20	9·5798	31	9·9661	6	9·6136	36	0·3864		40
		L. Cos.	Diff.	L. Sin.	Diff.	L. Cot.	Diff.	L. Tan.	Deg.	Min.

TABLE 25.

LOGARITHMS OF TRIGONOMETRICAL RATIOS (*Continued*).

Deg.	Min.	L. Sin.	Diff.	L. Cos.	Diff.	L. Tan.	Diff.	L. Cot.		
22	30	9·5828	31	9·9656	5	9·6172	36	0·3828	67	30
	40	9·5859	30	9·9651	5	9·6208	35	0·3792		20
	50	9·5889	30	9·9646	6	9·6243	36	0·3757		10
23	0	9·5919	29	9·9640	5	9·6279	35	0·3721	67	0
	10	9·5948	30	9·9635	6	9·6314	34	0·3686	66	50
	20	9·5978	29	9·9629	5	9·6348	35	0·3652		40
	30	9·6007	29	9·9624	6	9·6383	34	0·3617		30
	40	9·6036	29	9·9618	5	9·6417	35	0·3583		20
	50	9·6065	28	9·9613	6	9·6452	34	0·3548		10
24	0	9·6093	28	9·9607	5	9·6486	34	0·3514	66	0
	10	9·6121	28	9·9602	6	9·6520	33	0·3480	65	50
	20	9·6149	28	9·9596	6	9·6553	34	0·3447		40
	30	9·6177	28	9·9590	6	9·6587	33	0·3413		30
	40	9·6205	27	9·9584	5	9·6620	34	0·3380		20
	50	9·6232	27	9·9579	6	9·6654	33	0·3346		10
25	0	9·6259	27	9·9573	6	9·6687	33	0·3313	65	0
	10	9·6286	27	9·9567	6	9·6720	32	0·3280	64	50
	20	9·6313	27	9·9561	6	9·6752	33	0·3248		40
	30	9·6340	26	9·9555	6	9·6785	32	0·3215		30
	40	9·6366	26	9·9549	6	9·6817	33	0·3183		20
	50	9·6392	26	9·9543	6	9·6850	32	0·3150		10
26	0	9·6418	26	9·9537	7	9·6882	32	0·3118	64	0
	10	9·6444	26	9·9530	6	9·6914	32	0·3086	63	50
	20	9·6470	25	9·9524	6	9·6946	31	0·3054		40
	30	9·6495	26	9·9518	6	9·6977	32	0·3023		30
	40	9·6521	25	9·9512	7	9·7009	31	0·2991		20
	50	9·6546	24	9·9505	6	9·7040	32	0·2960		10
27	0	9·6570	25	9·9499	7	9·7072	31	0·2928	63	0
	10	9·6595	25	9·9492	6	9·7103	31	0·2897	62	50
	20	9·6620	24	9·9486	7	9·7134	31	0·2866		40
	30	9·6644	24	9·9479	6	9·7165	31	0·2835		30
	40	9·6668	24	9·9473	7	9·7196	30	0·2804		20
	50	9·6692	24	9·9466	7	9·7226	31	0·2774		10
28	0	9·6716	24	9·9459	6	9·7257	30	0·2743	62	0
	10	9·6740	23	9·9453	7	9·7287	30	0·2713	61	50
	20	9·6763	24	9·9446	7	9·7317	31	0·2683		40
	30	9·6787	23	9·9439	7	9 7348	30	0·2652		30
	40	9·6810	23	9·9432	7	9·7378	30	0·2622		20
	50	9·6833	23	9·9425	7	9·7408	30	0·2592		10
29	0	9·6856	22	9·9418	7	9·7438	29	0·2562	61	0
	10	9·6878	23	9·9411	7	9·7467	30	0·2533	60	50
	20	9·6901	22	9·9404	7	9·7497	29	0·2503		40
	30	9·6923	23	9·9397	7	9·7526	30	0·2474		30
	40	9·6946	22	9·9390	7	9·7556	29	0·2444		20
	50	9·6968		9·9383		9·7585		0·2415		10
		L. Cos.	Diff.	L. Sin.	Diff.	L. Cot.	Diff.	L. Tan.	Deg.	Min.

TABLE 25.

LOGARITHMS OF TRIGONOMETRICAL RATIOS (*Continued*).

Deg.	Min.	L. Sin.	Diff.	L. Cos.	Diff.	L. Tan.	Diff.	L. Cot.		
30	0	9·6990	22	9·9375	7	9·7614	30	0·2386	60	0
	10	9·7012	21	9·9368	7	9·7644	29	0·2356	59	50
	20	9·7033	22	9·9361	8	9·7673	28	0·2327		40
	30	9·7055	21	9·9353	7	9·7701	29	0·2299		30
	40	9·7076	21	9·9346	8	9·7730	29	0·2270		20
	50	9·7097	21	9·9338	7	9·7759	29	0·2241		10
31	0	9·7118	21	9·9331	8	9·7788	28	0·2212	59	0
	10	9·7139	21	9·9323	8	9·7816	29	0·2184	58	50
	20	9·7160	21	9·9315	7	9·7845	28	0·2155		40
	30	9·7181	20	9·9308	8	9·7873	29	0·2127		30
	40	9·7201	21	9·9300	8	9·7902	28	0·2098		20
	50	9·7222	20	9·9292	8	9·7930	28	0·2070		10
32	0	9·7242	20	9·9284	8	9·7958	28	0·2042	58	0
	10	9·7262	20	9·9276	8	9·7986	28	0·2014	57	50
	20	9·7282	20	9·9268	8	9·8014	28	0·1986		40
	30	9·7302	20	9·9260	8	9·8042	28	0·1958		30
	40	9·7322	20	9·9252	8	9·8070	27	0·1930		20
	50	9·7342	19	9·9244	8	9·8097	28	0·1903		10
33	0	9·7361	19	9·9236	8	9·8125	28	0·1875	57	0
	10	9·7380	20	9·9228	9	9·8153	27	0·1847	56	50
	20	9·7400	19	9·9219	8	9·8180	28	0·1820		40
	30	9·7419	19	9·9211	8	9·8208	27	0·1792		30
	40	9·7438	19	9·9203	9	9·8235	28	0·1765		20
	50	9·7457	19	9·9194	8	9·8263	27	0·1737		10
34	0	9·7476	18	9·9186	9	9·8290	27	0·1710	56	0
	10	9·7494	19	9·9177	8	9·8317	27	0·1683	55	50
	20	9·7513	18	9·9169	9	9·8344	27	0·1656		40
	30	9·7531	19	9·9160	9	9·8371	27	0·1629		30
	40	9·7550	18	9·9151	9	9·8398	27	0·1602		20
	50	9·7568	18	9·9142	8	9·8425	27	0·1575		10
35	0	9·7586	18	9·9134	9	9·8452	27	0·1548	55	0
	10	9·7604	18	9·9125	9	9·8479	27	0·1521	54	50
	20	9·7622	18	9·9116	9	9·8506	27	0·1494		40
	30	9·7640	17	9·9107	9	9·8533	26	0·1467		30
	40	9·7657	18	9·9098	9	9·8559	27	0·1441		20
	50	9·7675	17	9·9089	9	9·8586	27	0·1414		10
36	0	9·7692	18	9·9080	10	9·8613	26	0·1387	54	0
	10	9·7710	17	9·9070	9	9·8639	27	0·1361	53	50
	20	9·7727	17	9·9061	9	9·8666	26	0·1334		40
	30	9·7744	17	9·9052	10	9·8692	26	0·1308		30
	40	9·7761	17	9·9042	9	9·8718	27	0·1282		20
	50	9·7778	17	9·9033	10	9·8745	26	0·1255		10
37	0	9·7795	16	9·9028	9	9·8771	26	0·1229	53	0
	10	9·7811	17	9·9014	10	9·8797	27	0·1203	52	50
	20	9·7828		9·9004		9·8824		0·1176		40
		L. Cos.	Diff.	L. Sin.	Diff.	L. Cot.	Diff.	L. Tan.	Deg.	Min.

TABLE 25.

LOGARITHMS OF TRIGONOMETRICAL RATIOS (*Continued*).

Deg.	Min.	L. Sin.	Diff.	L. Cos.	Diff.	L. Tan.	Diff.	L. Cot.		
37	30	9·7844		9·8995		9·8850		0·1150	52	30
	40	9·7861	17	9·8985	10	9·8876	26	0·1124		20
	50	9·7877	16	9·8975	10	9·8902	26	0·1098		10
38	0	9·7893	16	9·8965	10	9·8928	26	0·1072	52	0
	10	9·7910	17	9·8955	10	9·8954	26	0·1046	51	50
	20	9·7926	16	9·8945	10	9·8980	26	0·1020		40
	30	9·7941	15	9·8935	10	9·9006	26	0·0994		30
	40	9·7957	16	9·8925	10	9·9032	26	0·0968		20
	50	9·7973	16	9·8915	10	9·9058	26	0·0942		10
39	0	9·7989	16	9·8905	10	9·9084	26	0·0916	51	0
	10	9·8004	15	9·8895	10	9·9110	26	0·0890	50	50
	20	9·8020	16	9·8884	11	9·9135	25	0·0865		40
	30	9·8035	15	9·8874	10	9·9161	26	0·0839		30
	40	9·8050	15	9·8864	10	9·9187	26	0·0813		20
	50	9·8066	16	9·8853	11	9·9212	25	0·0788		10
40	0	9·8081	15	9·8843	10	9·9238	26	0·0762	50	0
	10	9·8096	15	9·8832	11	9·9264	26	0·0736	49	50
	20	9·8111	15	9·8821	11	9·9289	25	0·0711		40
	30	9·8125	14	9·8810	11	9·9315	26	0·0685		30
	40	9·8140	15	9·8800	10	9·9341	26	0·0659		20
	50	9·8155	15	9·8789	11	9·9366	25	0·0634		10
41	0	9·8169	14	9·8778	11	9·9392	26	0·0608	49	0
	10	9·8184	15	9·8767	11	9·9417	25	0·0583	48	50
	20	9·8198	14	9·8756	11	9·9443	26	0·0557		40
	30	9·8213	15	9·8745	11	9·9468	25	0·0532		30
	40	9·8227	14	9·8733	12	9·9494	26	0·0506		20
	50	9·8241	14	9·8722	11	9·9519	25	0·0481		10
42	0	9·8255	14	9·8711	11	9·9544	25	0·0456	48	0
	10	9·8269	14	9·8699	12	9·9570	26	0·0430	47	50
	20	9·8283	14	9·8688	11	9·9595	25	0·0405		40
	30	9·8297	14	9·8676	12	9·9621	26	0·0379		30
	40	9·8311	14	9·8665	11	9·9646	25	0·0354		20
	50	9·8324	13	9·8653	12	9·9671	25	0·0329		10
43	0	9·8338	14	9·8641	12	9·9697	26	0·0303	47	0
	10	9·8351	13	9·8629	12	9·9722	25	0·0278	46	50
	20	9·8365	14	9·8618	11	9·9747	25	0·0253		40
	30	9·8378	13	9·8606	12	9·9772	25	0·0228		30
	40	9·8391	13	9·8594	12	9·9798	26	0·0202		20
	50	9·8405	14	9·8582	12	9·9823	25	0·0177		10
44	0	9·8418	13	9·8569	13	9·9848	25	0·0152	46	0
	10	9·8431	13	9·8557	12	9·9874	26	0·0126	45	50
	20	9·8444	13	9·8545	12	9·9899	25	0·0101		40
	30	9·8457	13	9·8532	13	9·9924	25	0·0076		30
	40	9·8469	12	9·8520	12	9·9949	25	0·0051		20
	50	9·8482	13	9·8507	13	9·9975	26	0·0025		10
45	0	9·8495	13	0·8495	12	0·0000	25	0·0000	45	0
		L. Cos.	Diff.	L. Sin.	Diff.	L. Cot.	Diff.	L. Tan.	Deg.	Min.

TABLE 26.

EXPONENTIAL AND HYPERBOLIC FUNCTIONS.

x	e^x	e^{-x}	Cosh x	Sinh x	x	e^x	e^{-x}	Cosh x	Sinh x
·00	1·0000	1·0000	1·0000	·0000	·40	1·4918	·6703	1·0811	·4108
·01	1·0100	·9900	1·0000	·0100	·41	1·5068	·6636	1·0852	·4216
·02	1·0202	·9802	1·0002	·0200	·42	1·5220	·6570	1·0895	·4325
·03	1·0304	·9704	1·0004	·0300	·43	1·5373	·6505	1·0939	·4434
·04	1·0408	·9608	1·0008	·0400	·44	1·5527	·6440	1·0984	·4543
·05	1·0513	·9512	1·0012	·0500	·45	1·5683	·6376	1·1030	·4653
·06	1·0618	·9418	1·0018	·0600	·46	1·5841	·6313	1·1077	·4764
·07	1·0725	·9324	1·0024	·0701	·47	1·6000	·6250	1·1125	·4875
·08	1·0833	·9231	1·0032	·0801	·48	1·6161	·6188	1·1174	·4986
·09	1·0942	·9139	1·0040	·0901	·49	1·6323	·6126	1·1225	·5098
·10	1·1052	·9048	1·0050	·1002	·50	1·6487	·6065	1·1276	·5211
·11	1·1163	·8958	1·0061	·1102	·51	1·6653	·6005	1·1329	·5324
·12	1·1275	·8869	1·0072	·1203	·52	1·6820	·5945	1·1383	·5438
·13	1·1388	·8781	1·0085	·1304	·53	1·6989	·5886	1·1438	·5552
·14	1·1503	·8694	1·0098	·1405	·54	1·7160	·5828	1·1494	·5666
·15	1·1618	·8607	1·0113	·1506	·55	1·7332	·5770	1·1551	·5781
·16	1·1735	·8521	1·0128	·1607	·56	1·7507	·5712	1·1609	·5897
·17	1·1853	·8437	1·0145	·1708	·57	1·7683	·5655	1·1669	·6014
·18	1·1972	·8353	1·0162	·1810	·58	1·7860	·5599	1·1730	·6131
·19	1·2092	·8270	1·0181	·1912	·59	1·8040	·5543	1·1792	·6248
·20	1·2214	·8187	1·0201	·2013	·60	1·8221	·5488	1·1855	·6366
·21	1·2337	·8106	1·0221	·2116	·61	1·8404	·5434	1·1919	·6485
·22	1·2461	·8025	1·0243	·2218	·62	1·8589	·5379	1·1984	·6605
·23	1·2586	·7945	1·0266	·2320	·63	1·8776	·5326	1·2051	·6725
·24	1·2712	·7866	1·0289	·2423	·64	1·8965	·5273	1·2119	·6846
·25	1·2840	·7788	1·0314	·2526	·65	1·9155	·5220	1·2188	·6968
·26	1·2969	·7710	1·0340	·2629	·66	1·9348	·5168	1·2258	·7090
·27	1·3100	·7634	1·0367	·2733	·67	1·9542	·5117	1·2330	·7213
·28	1·3231	·7558	1·0395	·2837	·68	1·9739	·5066	1·2402	·7336
·29	1·3364	·7483	1·0424	·2941	·69	1·9937	·5016	1·2476	·7461
·30	1·3499	·7408	1·0453	·3045	·70	2·0138	·4966	1·2552	·7586
·31	1·3634	·7334	1·0484	·3150	·71	2·0340	·4916	1·2628	·7712
·32	1·3771	·7262	1·0516	·3255	·72	2·0544	·4868	1·2706	·7838
·33	1·3910	·7189	1·0550	·3360	·73	2·0751	·4819	1·2785	·7966
·34	1·4050	·7118	1·0584	·3466	·74	2·0959	·4771	1·2865	·8094
·35	1·4191	·7047	1·0619	·3572	·75	2·1170	·4724	1·2947	·8223
·36	1·4333	·6977	1·0655	·3678	·76	2·1383	·4677	1·3030	·8353
·37	1·4477	·6907	1·0692	·3785	·77	2·1598	·4630	1·3114	·8484
·38	1·4623	·6839	1·0731	·3892	·78	2·1815	·4584	1·3199	·8615
·39	1·4770	·6771	1·0770	·4000	·79	2·2034	·4538	1·3286	·8748

TABLE 26 (*Continued*).

EXPONENTIAL AND HYPERBOLIC FUNCTIONS (*continued*).

x	e^x	e^{-x}	Cosh x	Sinh x	x	e^x	e^{-x}	Cosh x	Sinh
·80	2·2255	·4493	1·3374	·8881	1·20	3·3201	·3012	1·8107	1·5095
·81	2·2479	·4449	1·3464	·9015	1·21	3·3535	·2982	1·8258	1·5276
·82	2·2705	·4404	1·3555	·9150	1·22	3·3872	·2952	1·8412	1·5460
·83	2·2933	·4360	1·3647	·9286	1·23	3·4212	·2923	1·8568	1·5645
·84	2·3164	·4317	1·3740	·9423	1·24	3·4556	·2894	1·8725	1·5831
·85	2·3396	·4274	1·3835	·9561	1·25	3·4903	·2865	1·8884	1·6019
·86	2·3632	·4232	1·3932	·9700	1·26	3·5254	·2836	1·9045	1·6209
·87	2·3869	·4190	1·4029	·9840	1·27	3·5608	·2808	1·9208	1·6400
·88	2·4109	·4148	1·4128	·9981	1·28	3·5966	·2780	1·9373	1·6593
·89	2·4351	·4107	1·4229	1·0122	1·29	3·6328	·2753	1·9540	1·6788
·90	2·4596	·4066	1·4331	1·0265	1·30	3·6693	·2725	1·9709	1·6984
·91	2·4843	·4025	1·4434	1·0409	1·31	3·7062	·2698	1·9880	1·7182
·92	2·5093	·3985	1·4539	1·0554	1·32	3·7434	·2671	2·0053	1·7381
·93	2·5345	·3946	1·4645	1·0700	1·33	3·7810	·2645	2·0228	1·7583
·94	2·5600	·3906	1·4753	1·0847	1·34	3·8190	·2618	2·0404	1·7786
·95	2·5857	·3867	1·4862	1·0995	1·35	3·8574	·2592	2·0583	1·7991
·96	2·6117	·3829	1·4973	1·1144	1·36	3·8962	·2567	2·0764	1·8198
·97	2·6379	·3791	1·5085	1·1294	1·37	3·9354	·2541	2·0947	1·8406
·98	2·6645	·3753	1·5199	1·1446	1·38	3·9749	·2516	2·1132	1·8617
·99	2·6912	·3716	1·5314	1·1598	1·39	4·0148	·2491	2·1320	1·8829
1·00	2·7183	·3679	1·5431	1·1752	1·40	4·0552	·2466	2·1509	1·9043
1·01	2·7456	·3642	1·5549	1·1907	1·41	4·0960	·2441	2·1700	1·9259
1·02	2·7732	·3606	1·5669	1·2063	1·42	4·1371	·2417	2·1894	1·9477
1·03	2·8011	·3570	1·5790	1·2220	1·43	4·1787	·2393	2·2090	1·9697
1·04	2·8292	·3534	1·5913	1·2379	1·44	4·2207	·2369	2·2288	1·9919
1·05	2·8576	·3499	1·6038	1·2539	1·45	4·2631	·2346	2·2488	2·0143
1·06	2·8864	·3465	1·6164	1·2700	1·46	4·3060	·2322	2·2691	2·0369
1·07	2·9154	·3430	1·6292	1·2862	1·47	4·3492	·2299	2·2896	2·0596
1·08	2·9447	·3396	1·6421	1·3025	1·48	4·3930	·2276	2·3103	2·0826
1·09	2·9743	·3362	1·6552	1·3190	1·49	4·4371	·2254	2·3312	2·1059
1·10	3·0042	·3329	1·6685	1·3356	1·50	4·4817	·2231	2·3524	2·1293
1·11	3·0344	·3296	1·6820	1·3524	1·51	4·5267	·2209	2·3738	2·1529
1·12	3·0648	·3263	1·6958	1·3693	1·52	4·5722	·2187	2·3955	2·1768
1·13	3·0957	·3230	1·7093	1·3863	1·53	4·6182	·2165	2·4174	2·2008
1·14	3·1268	·3198	1·7233	1·4035	1·54	4·6646	·2144	2·4395	2·2251
1·15	3·1582	·3166	1·7374	1·4208	1·55	4·7115	·2122	2·4619	2·2496
1·16	3·1899	·3135	1·7517	1·4382	1·56	4·7588	·2101	2·4845	2·2743
1·17	3·2220	·3104	1·7662	1·4559	1·57	4·8066	·2080	2·5074	2·2993
1·18	3·2544	·3073	1·7808	1·4736	1·58	4·8550	·2060	2·5305	2·3245
1·19	3·2871	·3042	1·7956	1·4914	1·59	4·9038	·2039	2·5538	2·3499

TABLE 26. (*Continued*).

EXPONENTIAL AND HYPERBOLIC FUNCTIONS—*continued.*

x	e^x	e^{-x}	Cosh x	Sinh x	x	e^x	e^{-x}	Cosh x	Sinh x
1·60	4·9530	·2019	2·5775	2·3756	2·00	7·3891	·1353	3·7622	3·6269
1·61	5·0028	·1999	2·6014	2·4015	2·1	8·1662	·1225	4·1443	4·0219
1·62	5·0531	·1979	2·6255	2·4276	2·2	9·0250	·1108	4·5679	4·4571
1·63	5·1039	·1959	2·6499	2·4540	2·3	9·9742	·1003	5·0372	4·9370
1·64	5·1552	·1940	2·6746	2·4806	2·4	11·0232	·0907	5·5570	5·4662
1·65	5·2070	·1920	2·6995	2·5075	2·5	12·1825	·0821	6·1323	6·0502
1·66	5·2593	·1901	2·7247	2·5346	2·6	13·4637	·0743	6·7690	6·6947
1·67	5·3122	·1882	2·7502	2·5620	2·7	14·8797	·0672	7·4735	7·4063
1·68	5·3656	·1864	2·7760	2·5896	2·8	16·4446	·0608	8·2527	8·1919
1·69	5·4195	·1845	2·8020	2·6175	2·9	18·1741	·0550	9·1146	9·0596
1·70	5·4740	·1827	2·8283	2·6456	3·0	20·0855	·0498	10·0677	10·0179
1·71	5·5290	·1809	2·8549	2·6740	3·1	22·1980	·0450	11·1215	11·0765
1·72	5·5845	·1791	2·8818	2·7027	3·2	24·5325	·0408	12·2866	12·2459
1·73	5·6406	·1773	2·9090	2·7317	3·3	27·1126	·0369	13·5747	13·5379
1·74	5·6973	·1755	2·9364	2·7609	3·4	29·9641	·0334	14·9987	14·9654
1·75	5·7546	·1738	2·9642	2·7904	3·5	33·1155	·0302	16·5728	16·5426
1·76	5·8124	·1720	2·9922	2·8202	3·6	36·5982	·0273	18·3128	18·2855
1·77	5·8708	·1703	3·0206	2·8503	3·7	40·4473	·0247	20·2360	20·2113
1·78	5·9299	·1686	3·0492	2·8806	3·8	44·7012	·0224	22·3618	22·3394
1·79	5·9894	·1670	3·0782	2·9112	3·9	49·4024	·0202	24·7113	24·6911
1·80	6·0496	·1653	3·1075	2·9422	4·0	54·5982	·0183	27·3082	27·2899
1·81	6·1104	·1636	3·1370	2·9734	4·1	60·3403	·0166	30·1784	30·1619
1·82	6·1719	·1620	3·1669	3·0049	4·2	66·6863	·0150	33·3507	33·3357
1·83	6·2339	·1604	3·1972	3·0367	4·3	73·6998	·0136	36·8567	36·8431
1·84	6·2965	·1588	3·2277	3·0689	4·4	81·4509	·0123	40·7316	40·7193
1·85	6·3598	·1572	3·2585	3·1013	4·5	90·0171	·0111	45·0141	45·0030
1·86	6·4237	·1557	3·2897	3·1340	4·6	99·4843	·0100	49·7472	49·7371
1·87	6·4883	·1541	3·3212	3·1671	4·7	109·947	·0091	54·9781	54·9690
1·88	6·5535	·1526	3·3530	3·2005	4·8	121·510	·0082	60·7593	60·7511
1·89	6·6194	·1511	3·3852	3·2342	4·9	134·290	·0074	67·1486	67·1412
1·90	6·6859	·1496	3·4177	3·2682	5·0	148·413	·0067	74·2099	74·2032
1·91	6·7531	·1481	3·4506	3·3025	5·1	164·022	·0061	82·0140	82·0079
1·92	6·8210	·1466	3·4838	3·3372	5·2	181·272	·0055	90·6388	90·6333
1·93	6·8895	·1452	3·5173	3·3722	5·3	200·337	·0050	100·171	100·167
1·94	6·9588	·1437	3·5512	3·4075	5·4	221·406	·0045	110·705	110·701
1·95	7·0287	·1423	3·5855	3·4432	5·5	244·692	·0041	122·348	122·344
1·96	7·0093	·1409	3·6201	3·4792	5·6	270·426	·0037	135·215	135·211
1·97	7·1707	·1395	3·6551	3·5156	5·7	298·867	·0034	149·435	149·432
1·98	7·2427	·1381	3·6904	3·5523	5·8	330·300	·0030	165·151	165·148
1·99	7·3155	·1367	3·7261	3·5894	5·9	365·037	·0027	182·520	182·517
					6·0	403·429	·0025	201·716	201·713

SECTION II

STRENGTH OF MATERIALS AND STRUCTURES

STRENGTH OF MATERIALS AND STRUCTURES

Materials of Construction

IRON AND STEEL.

CAST IRON.

When iron ore is charged in a blast furnace, mixed with limestone as a flux, and melted down with charcoal, coke or anthracite as fuel, the resulting metal is run out into beds, and is known commercially as *pig iron.*

It contains about 73 per cent. of pure iron, about 3 to 5 per cent. of carbon, and some silicon, phosphorus, sulphur, etc.

The quality of cast iron suitable for forming castings varies according to the nature of the casting to be produced.

The following table of percentage composition is quoted from a paper by Mr. F. M. Thomas before the Birmingham Metallurgical Society—

TABLE 1.—COMPOSITION IRON CASTINGS.

Kind of Casting.	Combined Carbon.	Graphite Carbon.	Silicon.	Manganese.	Phosphorus.	Sulphur.
Cylinders and Hydraulic	·6–·75	2·5–2·8	1·0–1·3	·5–·7	·45–·7	·08–·12
Heavy Machinery	·45–·60	2·8–3·0	1·2–1·5	·6–·85	·4–·6	·06–·08
General Castings	·35–·45	3·0–3·2	2·0–2·25	1·0	1·0–1·3	·06–·08
Ornamental, thin	·08–·15	3·2–3·6	2·5–2·8	1·0	1·3–1·5	·03–·04

EFFECT OF VARIOUS INGREDIENTS ON CAST IRON.

Combined Carbon. This is a chemical compound of carbon with iron (iron carbide). It is very hard and brittle and makes the iron "white" in proportion to its amount.

Graphite Carbon. Graphite reduces shrinkage and is often desired for that purpose. For a given graphitic structure the silicon content should decrease as the thickness of metal increases. Graphite makes the casting relatively easy to machine but, if large flakes, liable to break under impulsive loads. When in the globular or star formations the iron is usually strong and tough.

Silicon. This is the next important constituent after carbon and iron, and exists chiefly as silicides of iron and manganese. Its chief effect is to encourage the formation of graphitic carbon at the expense of iron carbide, and thus to soften the casting.

Manganese. This is complex in its action. If in sufficient quantity it has a hardening effect, but its first action is to neutralise the hardening effect of sulphur and phosphorus. It tends also to form a double carbide of iron and manganese which is finer and stronger than the carbide of iron. It makes for soundness in castings but increases shrinkage.

Phosphorus. Phosphorus increases the fluidity of the iron and is thus useful for thin ornamental castings, but it is undesirable in quantities above about ·5 per cent. when strong castings are required. It discourages shrinkage but is liable to form strongly phosphoric pellets in spongy places.

Sulphur. This comes largely from the fuel which should be closely watched in this respect. It makes the iron sluggish in pouring, causes blow-holes, and increases shrinkage. The bad effects of sulphur can be neutralised in part by increasing the silicon and manganese content.

TABLE 2.—SHRINKAGE OF CASTINGS.

The usual allowance for each foot in length is as follows—

Large cast-iron cylinders ..	$\frac{3}{32}$ in.	Steel castings ..	$\frac{1}{4}$ in.
Small cast-iron cylinders ..	$\frac{1}{16}$ in.	Thick brass „ ..	$\frac{5}{32}$ in.
Cast-iron beams, etc. ..	$\frac{1}{10}$ in.	Thin „ „ ..	$\frac{3}{16}$ in.
Cast-iron pipe	$\frac{1}{8}$ in.	Aluminium „ ..	$\frac{1}{8}$ in.

MALLEABLE CAST IRON.

Malleable castings are produced by converting the combined carbon of white cast iron into an amorphous uncombined condition; this is done by heating the white cast iron to a temperature between 1380° F. and 2000° F. with an oxide of iron (usually hematite ore). The oxygen burns out part of the carbon and causes the material to resemble mild steel.

STEEL.

CLASSES OF STEEL.

The word "steel" is applied to many mixtures which differ greatly in chemical as well as in physical qualities. A comparatively small difference in the percentage of carbon in steel has a marked effect upon its quality. High-grade razor steel contains about 1·2 per cent. carbon, spring steel 1 per cent., steel rails ·5 per cent., and mild steel ·2 per cent.; the softest steel plate may contain as little as ·07 per cent. carbon.

Steel which is very low in carbon can easily be welded but it cannot be hardened; when the carbon content exceeds ·33 per cent. welding becomes more difficult. In tool steels other ingredients, such as nickel, manganese, tungsten, chromium, are sometimes used to vary the hardening qualities; in "high-speed" steels the hardness is not lost by cutting at high speed with resulting heating.

Steel was originally made by first obtaining practically pure iron from pig iron by means of the "puddling" process, and later heating with carbon by means of the "cementation" process.

Bessemer revolutionised the steel industry by producing steel direct from cast iron by burning out part of the carbon in a "converter" in which air was blown through the molten iron. After practically all the carbon and most of the impurtiies are burnt out a small amount of ferro-manganese is added, and the liquid steel is poured out into "ingot" moulds, the resulting ingots being rolled down into "blooms."

In the Siemens-Martin or open-hearth process the impurities are burnt out by placing the molten iron in a bath lying on the hearth of a regenerative furnace.

HARDENING, TEMPERING, AND ANNEALING STEEL.

Hardening steel is effected by heating the steel to a certain temperature and causing it to cool quickly—usually by quenching in water or oil.

Tempering is the process of removing some of the hardness and the accompanying brittleness of hardened steel, the usual process being to heat the hardened steel up to a certain temperature and allowing it to cool slowly.

Annealing is the process of reducing steel to what may be called its normal or soft condition.

HEAT TREATMENT OF STEEL.

Great advances have been made in recent years in the processes of hardening, tempering, and annealing steel by accurate investigation of the temperatures to which the heating should be conducted.

The past practice of judging temperatures by colours has proved unsatisfactory in cases where consistent results are required in bulk production and temperatures are now usually ascertained by pyrometer.

Critical Temperatures. If steel is heated uniformly and the increase in temperature is noted at equal time intervals, it will be noted that at a point called the *decalescence* point the steel absorbs heat without increase in temperature as indicated in the diagram on the next page.

The first point at which this occurs is called the first *critical* point; according to the quality of the steel there may be found one or two higher temperatures at which this occurs.

If the test be made by cooling, similar points called *recalescence* points are noted, the recalescence points occurring at slightly lower temperatures than the corresponding decalescence points.

These critical points bear some resemblance to the change-of-state points where ice changes into water and water into steam, the heat absorbed or given out corresponding to latent heat.

Steel in its normal form at ordinary temperatures has its carbon, which is the principal hardening element, in a certain form called *pearlite*, the remainder of the steel being practically pure iron called *ferrite*. The pearlite is a mechanical mixture of iron with iron carbide (Fe_3C), which is called *cementite*, and contains 6·6 per cent. of carbon; the percentage of carbon in pearlite is about ·9 per cent.

Now in passing through the critical temperature the pearlite is converted into an entirely new constituent with different properties called *Austenite*, and behaves just as if it were a "solid solution," and if the steel be suddenly cooled this new constituent remains and does not convert into pearlite.

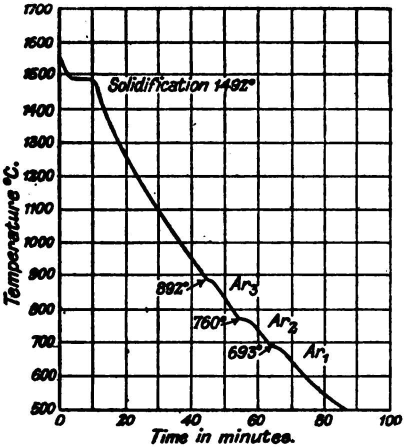


FIG. 1.—CRITICAL TEMPERATURES OF STEEL.

In order, therefore, to effect satisfactory hardening we should heat the steel to a temperature just above the critical temperature and then cool quickly.

Now the critical temperature varies for different qualities of steel and, as judgment of temperature by colour cannot be made closer than 100° F., it is clearly desirable to employ pyrometers in order to be confident that the correct temperature has been reached.

For complete annealing (sometimes referred to as *normalising*) the steel should be heated to a temperature just above the upper critical temperature.

For a full treatment of this important subject the reader is referred to a text-book by D. K. Bullens entitled "*Heat Treatment of Steel*" (Chapman & Hall, Ltd.).

Tempering Colours. For tool-making on a relatively small scale much of the work has to be done by relying on the tempering colours which appear upon the surface of brightened steel as heating proceeds; the following table may be used—

TABLE 3.—TEMPERATURE AND COLOUR OF STEEL.

° F.	Colour.	Nature of Work.
430	Very pale yellow	Milling cutters; hammer faces
440	Light yellow	Rotary profile cutters
450	Pale straw yellow	Twist drills, taps
460	Straw yellow	Plane irons; dies and punches
470	Deep straw yellow	Circular metal saws; wood bits
480	Dark yellow	Augers
490	Yellow-brown	Circular cutters for wood
500	Brown-yellow	Surgical instruments
510	Spotted red-brown	Axes
520	Brown purple	Snaps for pneumatic hammers
530	Light purple	Hack saws
540	Full purple	Smiths' tools
550	Dark purple	Screw drivers; wood saws
560	Full blue	Springs

NON-FERROUS METALS.

The non-ferrous metals are used in engineering practice principally in the form of alloys, particulars of some of which are given in the table on the next page. There are a very large number of proprietary alloys such as Delta metals, phosphor bronzes, manganese bronzes, and various brasses and white metals which are the productions of various firms and which are intended for particular purposes. The proportions used in the various alloys vary in practice according to the use to which they are to be put and to the experience of the firms supplying the metal.

Bronzes of various kinds are employed for manufacturing blanks for valve bodies and like articles by hot stamping or forging. Such stampings require very careful annealing, otherwise initial stresses may cause warping of the articles after machining.

Aluminium has a specific gravity of 2·71, and is one-third as heavy as iron or steel, and can be made to have a tensile strength about one-half that of mild steel.

Aluminium requires frequent annealing during working, and cannot easily be soldered or screwed; it does not readily tarnish but is easily attacked by alkali.

Aluminium is the principal constituent in a large number of light-weight non-rusting alloys, and has proved very effective in avoiding the formation of blow-holes in steel when added in small quantities in steel manufacture.

Powdered aluminium is employed in the alumino-thermic process (sometimes called *Thermit* process) of welding iron and steel. Powdered aluminium has a strong affinity for iron oxide, and if the two materials are brought together and ignited great heat results, and molten iron is deposited as indicated by the formula—

$$2\,Al + Fe_2\,O_3 = Al_2\,O_3 + 2\,Fe.$$

The aluminium oxide rises as a slag.

In practice the molten iron is either used to unite the steel or iron portions together, as in soldering or the parts are strongly clamped and the resulting heat and resisted expansion causes sufficient pressure to be generated to cause welding.

ALLOYS.

TABLE 4.—COMPOSITION OF COMMON ALLOYS.

Alloy.	Copper.	Zinc.	Tin.	Nickel.	Lead.	Antimony.	Bismuth.	Aluminium.
Babbitt's metal ..	3·7		89			7·3		
Brass, common ..	85	5	10					
,, hard	80	5	15					
,, white.. ..	10	80	10					
,, wire	67	33						
Britannia metal ..			75			25		
Bronze, aluminium	92·5							7·5
,, gilding metal	95		5					
,, gun metal ..	90		10					
,, statuary ..	91·4	5·5	1·4		1·7			
Cupro nickel.. ..	80			20				
German silver ..	33·3	33·4		33·3				
Muntz metal.. ..	60	40						
Pewter			80		20			
Solder, ordinary ..			66		34			
,, plumbers' ..			34		66			
Type metal					80	20		
White metal.. ..	2·4		54		33	10·6		
Duralumin ..								

Duralumin has been employed to a large extent in aircraft construction; its specific gravity varies between 2·77 and 2·84, and its melting point is about 650° C. It can be rolled, forged, and drawn both hot and cold, and can be screw-threaded. Its tensile strength varies from 50 to 65 thousands of lbs. per sq. inch, and its elongation percentage from 23 to 18, according to its composition.

Copper has a specific gravity of about 8·9 and is used principally for electrical purposes. It has low electric resistance, high thermal conductivity, and resists corrosion. It requires frequent annealing during working, but is very ductile in the annealed condition.

Copper is less easy to solder than most of its alloys; it is the principal constituent in the various brasses and bronzes which are classed as yellow metals.

Tin is a white malleable metal of specific gravity 7·3, and in its commercial form usually contains as impurities copper, iron, bismuth, and other metals. It melts at 450° F., and is thus often used as a safety plug for steam boilers. Its main commercial use is as a coating for iron sheets—called tin plates.

Zinc has a melting point of 780° F., and is used mainly for making brasses and for roofing, and for coating steel sheets (galvanizing).

TIMBER.

Timber is usually divided into two main classes—soft woods and hard woods.

Timber construction has been employed for centuries, and there are very many varieties of timber, the properties of the timber varying considerably even for the same variety according to the soil, time of growing, etc. Experience of many years in each country has shown certain kinds of timber to be useful for certain articles.

STANDARDS BY WHICH DEALS ARE SOLD.

Standard.	No. of pieces.	Size.	Cu. ft.	Ft. Super.
Petersburg	120	12 ft. × 11 ins. × 1½ ins.	165	1,320
London	120	12 ft. × 9 ins. × 3 ins.	270	1,080
Quebec				
(Long)	120	10 ft. × 11 ins. × 3 ins.	275	1,100
(Short)	100	12 ft. × 11 ins. × 2½ins.	229	1,100

The cheapness of the Baltic timbers in the past has not necessitated very careful attention on the part of engineers as to the strength, other considerations such as resistance to damp, ease of working by hand tools usually deciding the choice of material to be used.

RESISTANCE to CORROSION IN SEA-WATER.

This question has been very fully dealt with in the Report of the Institution of Civil Engineers (1920) on "Deterioration of Structures in Sea-water."

According to the summary of this report—

(*a*) At home ports, Baltic timber, which has been properly creosoted, would resist the attack of the teredo or leninoria to an extent which would justify the adoption of such treatment.

(*b*) The life of sea structure in connection with home ports would be further prolonged by the use of greenheart and to some extent by the employment of jarrah, and other Australian gums, in lieu of creosoted timber.

(*c*) In tropical waters, where the teredo and leninoria are much more active than around the British coasts, the use of creosote as a protective or preventive would not be justified by the results produced.

MATERIAL OF CONSTRUCTIONS.

CONCRETE.

Artificial stone or concrete is made by thoroughly mixing Portland cement with broken stone, wetting the mixture, and placing it in position whereupon it is allowed to set.

The broken stone is usually graded into two parts; that passing through a ¼-inch square mesh sieve is called the *sand* or fine material, and that not passing is called the *aggregate* or coarse material. For reinforced concrete work it is usual to specify that all material shall pass through a ¾-inch mesh square sieve.

The coarse material may consist of crushed rock, gravel or broken brick; the sand may be coarse river or pit sand, or small parts of broken stone.

Proportion of Materials. Concrete is usually specified by ratio of volumes of the component materials; thus 1 · 2 · 4 concrete means concrete comprising one volume of Portland cement to two volumes of sand and four volumes of coarse material.

The cement is the most expensive part of the concrete and to obtain consistently good strength with a minimum of strength requires careful attention to the proportion of the materials. The best proportions to be used cannot be laid down arbitrarily because they depend on the materials, although 1 · 2 · 4 is the mix most commonly employed in reinforced concrete work.

The best proportions are those which will give the greatest density of the mix; this can be found experimentally by weighing a pailful with varying proportions and ascertaining which weighs the most.

Mixing and Water Content. The materials must be thoroughly mixed before any water is added until the mixture is of an even colour throughout. In hand mixing a common rule is to turn over "three times dry and three times wet." The amount of water added is important; if more water is used than can be absorbed in the setting of the concrete it will utimately dry out, leaving voids in the concrete. Also if the concrete is mixed into a sloppy condition some of the cement will flow away through cracks in the moulds. It is difficult to lay down hard-and-fast rules as to consistency of concrete; with reinforced concrete the concrete must be sufficiently fluid to find its way in between the reinforcement but workmen are always likely to err on the side of making the concrete too wet. As a rough practical test, we may say that the concrete should be sufficiently wet that when hit with the back of a spade it will retain the impression of the spade.

It is important, particularly in hot weather, to prevent the concrete from drying out before the setting is complete.

Strength of Concrete. Concrete made with sharp sand and hard stone in the proportions of cement to sand to stone by volume 1 · 2 · 4, should develop a crushing strength of 1,600 lbs. per sq. inch in one month, and 2,400 lbs. per sq. inch in four months when tested upon cubes 4 ins. each way or cylinders 6 ins. each way. For designing, a working stress of 600 lbs. per sq. inch is usually allowed.

The tensile strength of concrete is approximately one-tenth of its compressive strength, but steel reinforcement is usually provided to resist the whole of the tensile forces.

For diagonal tensile stresses induced by shear it is usual to allow 60 lbs. per sq. inch in the concrete of the above mixture.

STRENGTH AND ELASTICITY OF MATERIALS.

STRAIN.

When forces act upon a body a change in size or shape of the body results, this change being called *strain*.

Strain may be of three principal kinds:—extension (tensile) contraction (compression), or slide (shear), and any form of strain occurring in three planes can be divided up into combinations of these forms.

In engineering practice we are usually concerned with intensity of strain or unit strain; in the case of extension this is equal to the extension of the body divided by its original length.

STRESS.

When an elastic body becomes strained, molecular forces in the material are induced which tend to bring the body back to its original form ; these molecular forces are called stresses and corresponding to the three kinds of strain we have tensile, compressive, and shear stresses.

In practice we are usually concerned with the *intensity of stress*, i.e. the molecular force acting over a unit area.

The stresses in a body correspond to the reaction caused by the external force acting, the Newtonian principle holding that to every action there is an equal and opposite reaction.

ELASTIC MODULUS.

For a perfectly elastic material intensity of stress is proportional to unit strain, and the ratio between the two is called an *elastic modulus*.

In compression or tension the modulus is known as Young's modulus (E) and in shear it is known as the Glide or rigidity modulus (G).

In tension or compression, therefore, we can calculate the strain when the stress is given by means of the relation

$$\text{Unit strain} = \frac{\text{Intensity of stress}}{\text{Young's modulus}}$$

POISSON'S RATIO.

If a material is stretched in one direction it contracts in a direction at right angles, and if it is compressed in one direction it stretches in a direction at right angles. The relation between the unit direct strain and the unit transverse strain is known as Poisson's ratio (η). We therefore have

$$\text{Poisson's ratio} = \eta = \frac{\text{Transverse unit strain}}{\text{Direct unit strain}}$$

η = approx. $\frac{1}{4}$ for most elastic materials.

Ratio between Elastic Moduli. It can be shown that $\frac{E}{G} = 2\,(1 + \eta)$. G therefore is usually $\frac{3}{8}$ E.

STRESS-STRAIN DIAGRAMS.

If a diagram of strains be plotted against stresses in a test of a ductile specimen the resulting diagram will be found at first to be a straight line until a certain stress called the *elastic limit* is reached beyond which the strain increases more quickly than the stress. At a stress slightly in excess of this elastic limit there is a sudden increase of strain, and the metal begins to flow, this point being called the *yield point.* It has been shown that in ductile materials this yield is due to slipping of the crystals of the material so that failure in tension is in reality due to indirect shear.

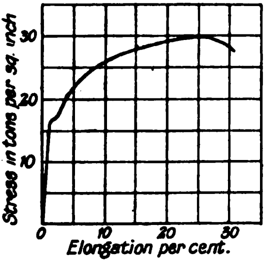


FIG. 2.—STRESS-STRAIN DIAGRAM.

As the stress increases the material continues to draw out until the ultimate stress is reached and the specimen draws down locally and then breaks.

In many stress diagrams there is shown an apparent diminution in strength after the alternate strength is reached. This diminution is apparent only and is due to the fact that the area of the specimen has been reduced; if the load be divided by this reduced area and a corrected stress be calculated it will be found that the stress continues right up to fracture.

DUCTILITY.

This is the quality of a material upon which depends the power of the material to stretch without cracking. Ductility is a property opposed to brittleness, and is a measure of the capacity to resist shocks. Ductility is usually measured by the percentage extension on fracture by tension or by the percentage reduction in area on fracture by tension. Many authorities maintain that reduction of area is the more reliable criterion although it is not so commonly specified.

Cold bend and flattening tests are often made to test ductility.

HARDNESS OF A METAL.

This is a quality that is not easy to measure satisfactorily, particularly in very hard metals, where a "skin" hardness is necessary. Two methods of testing are now most commonly employed.

(*a*) *Brinell Test.* In this test a hardened steel ball is pressed with a predetermined force against a plate whose hardness is required; the diameter of the resulting curved depression is then found and from this the "hardness number" is obtained in the following manner—

If P is the load, D the diameter of the ball, and d that of the impression, the quantity

$$H = \frac{P}{\frac{\pi D}{2}\left(D - \sqrt{D^2 - d^2}\right)}$$

is called the Brinell Hardness Number. The table on page 144 gives values.

(*b*) *Shore's Scleroscope Test.* In this test a steel plunger is released from a given height and falls on to the surface the hardness of whch is to be measured and the height of rebound is noted. The reading gives the Shore number, and a block of standard hardness is supplied with the instrument for checking the calibration.

The instrument is very neatly designed and with care quite good results can be obtained; this test has the advantage that it can be made as specimens without deforming them in any way.

Relation between Hardness and Strength of Carbon Steels. The tensile strength may be found approximately when the hardness is known, from the following formulae—

$$T = \cdot 73B - 28$$
$$T = 4{\cdot}4S - 28$$

where T = Tensile strength in thousands of pounds per sq. in.
B = Brinell Hardness Number
S = Shore Hardness Number

TABLE 5.—BRINELL'S HARDNESS NUMBERS (for Load 3,000 kg.).

Diameter of Steel Ball = 10 mm.

Diameter of Ball Impression.	Hardness No.	Diameter of Ball Impression.	Hardness No.	Diameter of Ball Impression.	Hardness No.	Diameter of Ball Impression.	Hardness No.
mm.		mm.		mm.		mm.	
2·0	946	3·25	351	4·50	179	5·75	105
2·05	898	3·30	340	4·55	174	5·80	103
2·10	857	3·35	332	4·60	170	5·85	101
2·15	817	3·40	321	4·65	166	5·90	99
2·20	782	3·45	311	4·70	163	5·95	97
2·25	744	3·50	302	4·75	159	6·0	95
2·30	713	3·55	293	4·80	156	6·05	94
2·35	683	3·60	286	4·85	153	6·10	92
2·40	652	3·65	277	4·90	149	6·15	90
2·45	627	3·70	269	4·95	146	6·20	89
2·50	600	3·75	262	5·0	143	6·25	87
2·55	578	3·80	255	5·05	140	6·30	86
2·60	555	3·85	248	5·10	137	6·35	84
2·65	532	3·90	241	5·15	134	6·40	82
2·70	512	3·95	235	5·20	131	6·45	81
2·75	495	4·0	228	5·25	128	6·50	80
2·80	477	4·05	223	5·30	126	6·55	79
2·85	460	4·10	217	5·35	124	6·60	77
2·90	444	4·15	212	5·40	121	6·65	76
2·95	430	4·20	207	5·45	118	6·70	74
3·0	418	4·25	202	5·50	116	6·75	73
3·05	402	4·30	196	5·55	114	6·80	71·5
3·10	387	4·35	192	5·60	112	6·85	70
3·15	375	4·40	187	5·65	109	6·90	69
3·20	364	4·45	183	5·70	107	6·95	68

For other test loads, the hardness numbers are proportional to those in the table.

STRENGTH OF MATERIALS.

(Average Values.)

(The bending stress given is that obtained with rectangular sections by dividing the bending moment at fracture by the section modulus.)

The ultimate compressive strength of ductile materials is difficult to measure exactly because such metals flow under heavy loads; in brittle materials such as stone and concrete the compressive strength depends upon the material used to transmit the load from the platen of the testing machine to the test block.

TABLE 6.—STRENGTH OF MATERIALS (METALS).

MATERIAL : METALS.	All figures in thousands of pounds (kips) per sq. inch.					
	Ultimate or Breaking Stresses				Elastic Limit in Tension.	Young's Modulus E.
	Tension.	Compression.	Shear.	Bending.		
Aluminium castings	15	12	12		6·5	11,000
„ sheet .	25				12·5	
Brass, cast . .	25	40	36	16	6·5	9,000
„ wire (hard) .	80					14,000
Bronze, phosphor .	50				24	
Copper bolts .	36					
„ cast . .	25	40	30	22	6	10,000
„ sheet .	31				10	
„ wire (hard)	60					18,000
„ (annealed)	35					15,000
Delta metal, cast .	55					
„ extruded bar	82					
Gun metal (tin 1, copper 9)	36			52	10	10,000
Iron, cast . .	15	80	18	30	6	12,000
„ malleable, cast	31				20	
„ wrought bars	50	48	40	54	27	28,000
„ wire (hard) .	80					
„ „ (annealed)	60					
Steel, cast . .	70	70	60	70	40	30,000
„ structural .	67	58	50	70	36	30,000
„ nickel (annealed)	80				48	30,000
„ wire (hard) .	120				80	30,000
„ „ (annealed)	80				60	30,000

To obtain the working stresses to be adopted in design the above figures should be divided by the *factor of safety*, which for static loading is usually taken as 4.

STRESSES IN BEAMS.

The ordinary theory of the stresses in beams is based upon the following assumptions—

(1) That a section of the beam which is plane before bending remains plane after bending (Bemoulli's assumption).

(2) That stress is proportional to strain for the material and that Young's modulus has the same value in tension as in compression.

(3) That the initial curvature of the beam is negligible.

This results in the following formulae

$$\frac{B}{I} = \frac{c}{n} = \frac{t}{(d-n)} = \frac{E}{R} \qquad (1)$$

where B = bending moment on the beam ;
I = moment of inertia of beam section about the neutral axis ;
c = maximum compression stress in the material ;
t = „ tension „ „ „
n = distance from compression edge to neutral axis ;
d = depth of beam ;
E = Young's modulus of the material of the beam ;
R = radius of beam after bending.

TABLE 7.—STRENGTH OF MATERIALS (NON-METALLIC).

MATERIAL :	All figures in thousands of pounds (kips) per sq. in.				
	Ultimate or breaking stresses.				Young's Modulus E.
TIMBERS.	Tension (with Grain for Timbers)	Compression.	Shear (with Grain for Timbers)	Bending.	
Ash	12	6		10	1,600
Beech	16	8		10	1,400
Birch	15	6		12	
Cedar (W. Indian)	5	6			
Chestnut	9	5	·6	5	1,200
Deal	12	6		9	1,600
Elm, English	13	10		6	1,300
Larch	9	4		9	1,000
Mahogany, Honduras	21	8		12	1,600
„ Spanish	21	8		8	1,600
Oak, American	10	6	·8	7	1,400
„ English	15	8		10	1,400
Pine, pitch	12	5		7	1,900
„ red	13	6		8	1,800
„ white	9	5		7	1,100
„ yellow	10	6		8	1,500
Teak	12	12		12	2,400
CEMENTS, STONES, &C.					
Brick, common		1·0			
„ London stock		2·5			
„ Stafford blue		6·0			
Cement, Portland (28 days)		·54			
Concrete, brick or gravel (1/2/4)		2·4			
Glass, green	4	30		4	
Flint	3	25		3	
Granite		15			
Limestone		7			
Marble		10			
Sandstone		8			

Taking the first three parts of the equation this gives,

$$B = \frac{cI}{n} = \frac{tI}{(d - n)} \qquad (2)$$

The quantities $\frac{I}{n}$ and $\frac{I}{(d \quad n)}$ are called respectively the *compression and tension moduli* of the section, and are given the symbols M_C and M_T.

In many beams the sections are symmetrical about the neutral axis so that $n = (d - n)$; in such case we have only one section modulus to deal with.

(*See* pages 180 and 181 for moduli of standard sections.)

TORSION OF SHAFTS.

The theory of shafting gives

$$\frac{T}{I_p} = \frac{s}{r} = \frac{G\theta}{l}$$

where T = twisting moment on shaft;
I_p = polar moment of inertia of shaft
s = maximum shear stress in shaft
r = radius of shaft;
G = glide or rigidity modulus of the material
θ = angle of twist of shaft;
l = length of shaft.

For a solid shaft of diameter D, this gives

$$T = s\frac{\pi D^3}{16} = \cdot 196\ s\ D^3$$

For a hollow shaft of external diameter D and internal diameter d we have

$$T = \frac{s\pi\ (D^4 - d^4)}{16\ D}$$

Horse-power Transmitted by Solid Shafting

$$D = 5{\cdot}23\sqrt[3]{\frac{H - P}{Ns}}$$

where D = diameter in inches;
HP = horse-power transmitted;
N = number of revolutions per minute
s = maximum shear stress in tons per sq. in.

(For a table for horse-power transmitted by steel shafting *see* p. 149.)

COMBINED TWISTING AND BENDING IN SHAFTS.

If a shaft transmit a twisting moment T and a bending moment B, it should be designed to carry an equivalent twisting moment T_E given by the formula

$$T_E = \sqrt{B^2 + T^2}$$

TORSION OF NON-CIRCULAR SHAFTS.

The following formulae due to St. Venant may be employed—

TABLE 8.

Section.	Relation of Maximum Stress to Twisting Moment.	Angles of Torsion in Degrees.
Circular. Dia. = D.	$T = s.\ \frac{\pi\ D\ d^2}{16}$	$\theta = \frac{292\ T\ l\ (d^2 + D^2)}{G\ D^3\ d^3}$
Square. Length of Side = S.	$T = s\ .\ {\cdot}208\ S^3$	$\theta = \frac{410\ T l}{G\ S^4}$
Rectangular. Sides B and H.	$T = \frac{s\ B\ H^2}{3 + 1{\cdot}8\ \frac{H}{B}}$	$\theta = \frac{205\ T\ l\ (B^2 + H^2)}{G\ B^3\ H^3}$
Any symmetrical section not containing re-entrant angles.	$T = \frac{s A^4}{40\ I_p . y}$ Where A = area of section I_p = polar moment of Inertia y = distance of farthest edge from centre of shaft	$\theta = \frac{40\ I_p T\ l}{A^4\ G}$

TABLE 9.—HORSE-POWER TRANSMITTED BY STEEL SHAFTING.

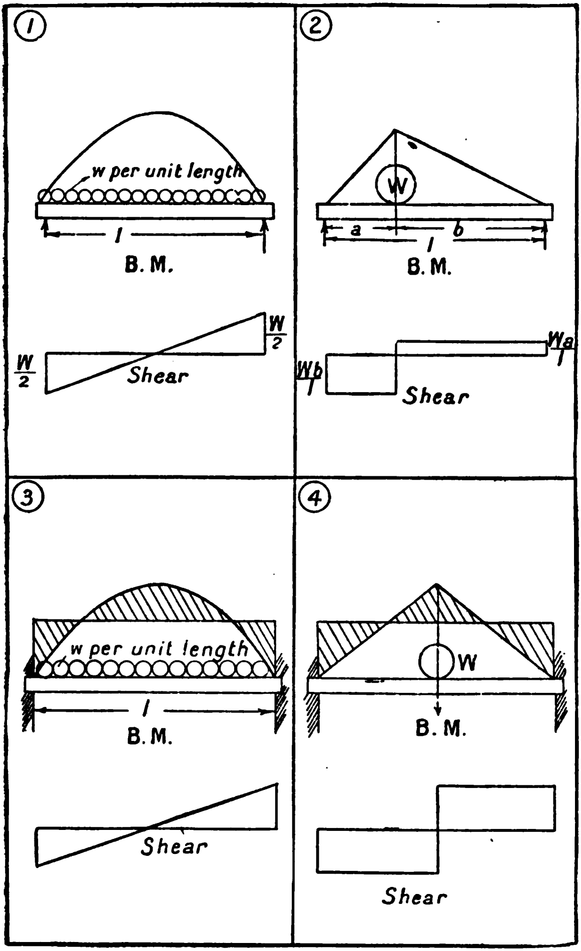
GIRDERS AND BEAMS.

Revolutions per minute.	DIAMETER OF SHAFT IN INCHES.																		
	1½	1¾	2	2¼	2½	2¾	3	3½	4	4½	5	5½	6	6½	7	7½	8	9	10
50	3·3	5·3	8·0	10·9	15·6	20·8	27	43	64	91	125	166	216	275	343	422	512	729	1000
60	4·0	6·4	9·6	13·1	18·8	25·0	32	51	77	109	150	200	259	330	412	506	614	875	1200
70	4·7	7·5	11·2	15·2	21·9	29·1	38	60	89	128	175	233	302	385	480	591	717	1021	1400
80	5·4	8·5	12·8	17·4	25·0	33·3	43	69	102	146	200	266	346	439	549	675	819	1166	1600
90	6·0	9·6	14·4	19·6	28·1	37·4	49	77	115	164	225	299	389	494	617	759	922	1312	1800
100	6·7	10·7	16·0	21·8	31·2	41·6	54	86	128	182	250	333	432	549	686	844	1024	1458	2000
110	7·4	11·8	17·6	23·9	34·4	45·8	59	94	141	200	275	366	475	604	755	928	1126	1604	2200
120	8·1	12·9	19·2	26·1	37·5	49·9	65	103	154	219	300	399	518	659	823	1012	1229	1750	2400
130	8·7	13·9	20·8	28·3	40·6	54·1	70	111	166	237	325	433	562	714	892	1097	1331	1895	2600
140	9·4	15·0	22·4	30·5	43·8	58·2	76	120	179	255	350	466	605	769	960	1181	1434	2041	2800
150	10·1	16·1	24·0	32·6	46·9	62·4	81	129	192	273	375	499	648	824	1029	1265	1536	2187	3000
160	10·8	17·1	25·6	34·8	50·0	66·5	86	137	205	292	400	532	691	879	1097	1350	1638	2333	3200
170	11·5	18·2	27·2	37·0	53·1	70·7	92	146	218	310	425	566	734	934	1166	1434	1741	2479	3400
180	12·2	19·3	28·8	39·2	56·3	74·9	97	154	230	328	450	599	778	989	1235	1519	1843	2624	3600
190	12·8	20·4	30·4	41·3	59·4	79·0	103	163	243	346	475	632	821	1044	1303	1603	1945	2770	3800
200	13·5	21·4	32·0	43·5	62·5	83·2	108	172	256	365	500	665	864	1099	1372	1687	2048	2916	4000
225	15·2	24·1	36·0	49·0	70·3	93·6	122	193	288	410	563	749	972	1236	1543	1898	2304	3280	4500
250	16·9	26·8	40·0	54·4	78·1	104·0	135	214	320	456	625	832	1080	1373	1715	2109	2560	3645	5000
275	18·6	29·5	44·0	59·8	85·9	114·4	149	236	352	501	688	915	1188	1510	1886	2320	2816	4009	5500
300	20·3	32·2	48·0	65·3	93·7	124·8	162	257	384	547	750	998	1296	1648	2058	2531	3072	4374	6000
325	21·9	34·8	52·0	70·7	101·6	135·2	176	279	416	592	813	1081	1404	1785	2229	2742	3328	4739	6500
350	23·6	37·5	56·0	76·2	109·4	145·6	189	300	448	638	875	1165	1512	1922	2401	2953	3584	5103	7000
375	25·3	40·2	60·0	81·6	117·2	156·0	203	322	480	683	938	1248	1620	2060	2572	3164	3840	5468	7500
400	27·0	42·9	64·0	87·0	125·0	166·4	216	343	512	729	1000	1331	1728	2197	2744	3374	4096	5832	8000
425	28·7	45·6	68·0	92·5	132·8	176·8	230	364	544	775	1063	1414	1836	2334	2915	3585	4352	6197	8500
450	30·4	48·2	72·0	97·9	140·6	187·2	243	386	576	820	1125	1497	1944	2472	3087	3796	4608	6562	9000
475	32·1	50·9	76·0	103·4	148·4	197·6	257	407	603	866	1188	1580	2052	2609	3258	4007	4864	6926	9500
500	33·7	53·6	80·0	108·8	156·2	208·0	270	429	640	911	1250	1684	2160	2746	3430	4218	5120	7290	10000

TABLE 10.—STANDARD CASES OF BENDING MOMENTS, SHEARING FORCES, AND DEFLECTIONS.

FIXED LOADS.

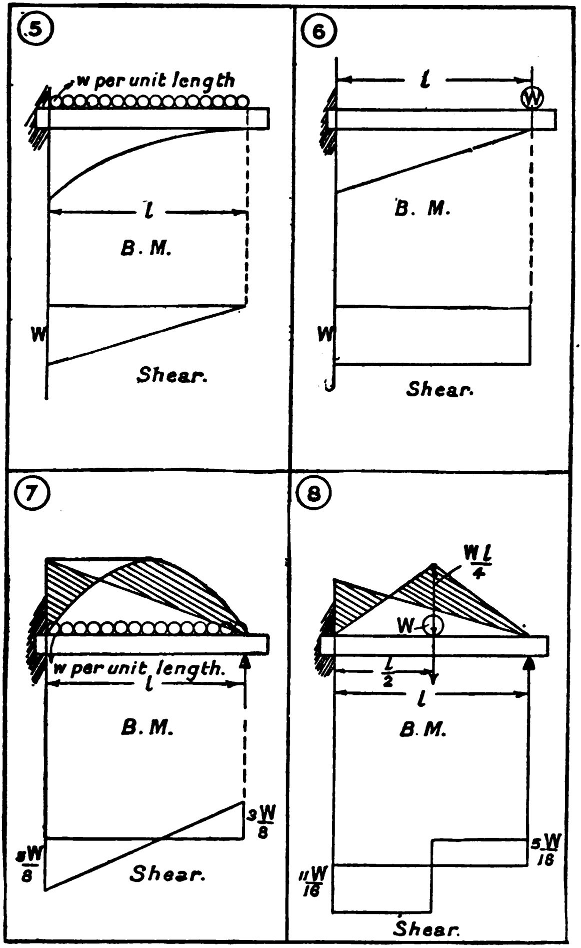
Case.	Bending Moment.	Shearing Force.	Deflection.
I Uniform load Simply supported	$B_c = \frac{wl^2}{8} = \frac{Wl}{8}$ $B_x = \frac{wx}{2}(l-x)$	S at each end $= \frac{W}{2} = \frac{wl}{2}$	Max. $d_n = \frac{5Wl^3}{384EI}$
2 Isolated load Simply supported	$B_{max} = \frac{Wab}{l}$ If load is central $B_c = \frac{Wl}{4}$	S at left end $= \frac{Wb}{l}$ S at right end $= \frac{Wa}{l}$ If load is central S each end $= \frac{W}{2}$	Max d_n $= \frac{Wab(2l-a)}{9EIl}\sqrt{\frac{a}{3}(2l-a)}$ If load is central Max. $d_n = \frac{Wl^3}{48EI}$
3 Uniform load Ends fixed	$B_c = \frac{wl^2}{24} = \frac{Wl}{24}$ $B_E = \frac{wl^2}{12} = \frac{Wl}{12}$	Same as 1	Max. $d_n = \frac{Wl^3}{384EI}$
4 Central load Ends fixed	$B_c = \frac{Wl}{8}$ $B_E = \frac{Wl}{8}$	$\frac{W}{2}$ at each end	Max. $d_n = \frac{Wl^3}{192EI}$



STANDARD CASES OF BEAMS—FIXED LOADS.

TABLE 10 (*contd.*).—BENDING MOMENTS, SHEARING FORCES, AND DEFLECTIONS
FIXED LOADS.

Case.	Bending Moment.	Shearing Force.	Deflection.
5 Uniform load Cantilever	$B_E = \frac{Wl}{2} = \frac{wl^2}{2}$ $B_X = \frac{W}{2}\left(l - x\right)^2$	$S = O$ at free end $= W$ at fixed end	d_n max. $= \frac{Wl^3}{8EI}$
6 Isolated load Cantilever	$B_E = Wl$ $B_X = W\ (l-x)$	$S = W$ at each end	d_n max. $= \frac{Wl^3}{3EI}$
7 Uniform load Beam fixed at one end; freely supported at other	$B_E = \frac{wl^2}{8} = \frac{Wl}{8}$ Max. B.M. on right side $= \frac{9wl^2}{128}$ Height of parabola $= \frac{Wl}{8}$	S at fixed end $= \frac{5Wl}{8}$ S at freely supported end $= \frac{3Wl}{8}$	d_n max. $= \frac{\cdot 0054Wl^3}{EI}$ at $\cdot 578l$ from fixed end
8 Central load Beam fixed at one end; freely supported at other	$B_E = \frac{3Wl}{16}$ $B_X = \frac{Wl}{4} - \frac{3Wl}{32}$ $= \frac{5Wl}{32}$	S at fixed end $= \frac{11W}{16}$ S at freely supported end $= \frac{5W}{16}$	d_n max. $= \frac{7Wl^3}{768EI}$

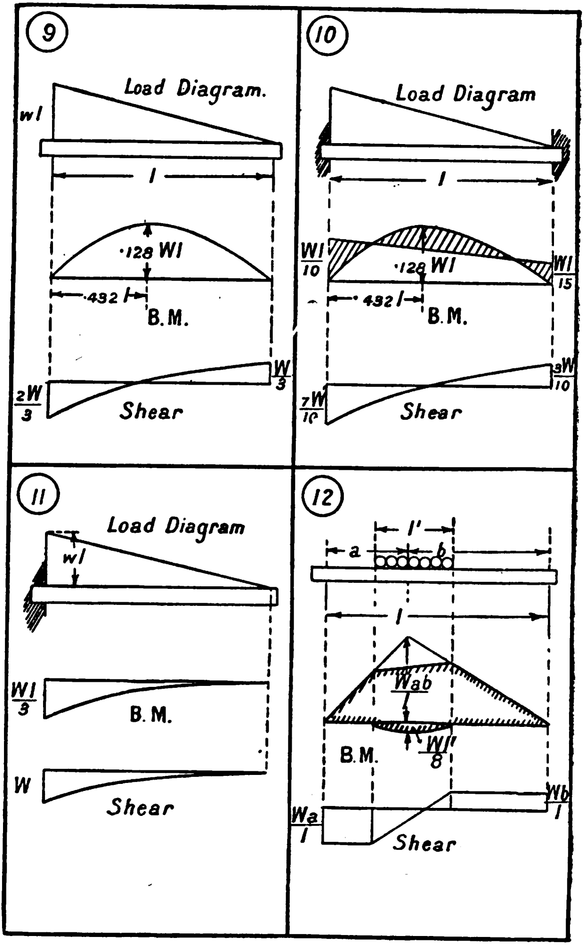


STANDARD CASES OF BEAMS—FIXED LOADS.

TABLE 10 (*contd.*).—BENDING MOMENTS, SHEARING FORCES, AND DEFLECTIONS.

FIXED LOADS.

Case.	Bending Moment.	Shearing Force.	Deflection.
9 Load gradually increasing in intensity from zero to max. Ends freely supported	B max. = ·128Wl at dist. ·423l from more heavily loaded end	S at heavily loaded end = $\frac{2W}{3}$ S at other end = $\frac{W}{3}$ $W = \frac{wl^2}{2}$	d_n max. = ·013 $\frac{Wl^3}{EI}$ at ·481l from heavily loaded end
10 Load as in 9 Ends fixed	$B_E = \frac{Wl}{10}$ B at other end = $\frac{Wl}{15}$ Outline of B.M. diagram same as previous case	S at heavily loaded end = $\frac{7W}{10}$ S at other end = $\frac{3W}{10}$ $W = \frac{wl^2}{2}$	—
11 Load as in 9 Cantilever fixed at more heavily loaded end	$B_E = \frac{Wl}{3}$ Curve is a parabola of the 3rd order $B_X = \frac{W(l-x)^3}{6}$	S at heavily loaded end $= W = \frac{wl^2}{2}$ At other end S = O	d_n max. at free end $= \frac{Wl^3}{15EI}$
12 Uniform load of length less than span. Ends freely supported	B.M. diagram as shown shaded	S at left hand end = $\frac{Wa}{l}$ S at other end = $\frac{Wb}{l}$	—

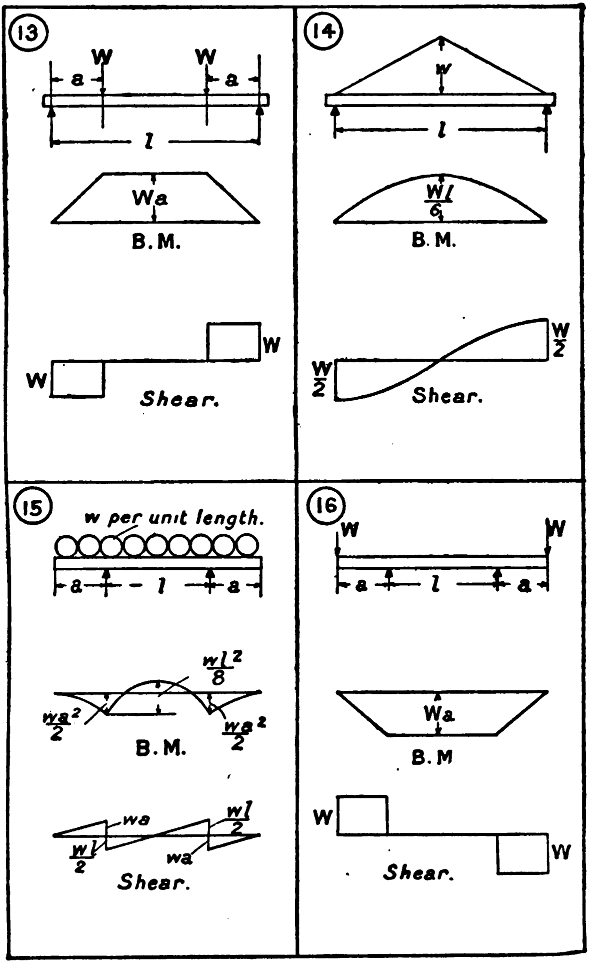


STANDARD CASES OF BEAMS—FIXED LOADS.

TABLE 10 (*contd.*).—BENDING MOMENTS, SHEARING FORCES, AND DEFLECTIONS.

FIXED LOADS.

Case.	Bending Moment.	Shear.	Deflection.
13 Two isolated loads equidistant from ends Simply supported	B_c = Wa between loads B_x = Wx until $x = a$	S = W up to loads S = O between loads	Max. $d_n = \frac{Wa}{24EI}\left(3l^2 - 4a^2\right)$ for $a = \frac{l}{3}$ $= \frac{Wl^3}{28{\cdot}2EI}$
14 Triangular load Simply supported	$B_c = \frac{Wl}{6}$ $B_x = W\left(\frac{x}{2} - \frac{2x^3}{3l^2}\right)$	$S = \frac{W}{2}$ at ends $S_c = O$ Curves are parabolas	Max. $d_n = \frac{Wl^3}{60EI}$
15 Uniformly distributed load Ends overhanging equally	B at supports $= \frac{Wa^2}{2}$ $B_c = W\left(\frac{l^2}{8} - \frac{a}{2}\right)$	S at supports $= Wa$ and $\frac{Wl}{2}$ (opposite in sign)	d_n at ends $= \frac{Wa_4}{8EI}$ d_n at centre $= \frac{Wl^2}{EI}\left\}\frac{5l^2}{384} - \frac{a_2}{16}\right\{$
16 Isolated loads at ends. Ends overhanging equally	B = Wa between supports B_x = Wx until $x = a$	S = W up to supports = O between loads	d_n at ends $= \frac{Wa_3}{3EI}$ d_n at centre $= \frac{-Wal^2}{8EI}$



STANDARD CASES OF BEAMS—FIXED LOADS.

CONTINUOUS BEAMS.

When a beam rests on more than two supports it is said to be "continuous." In this case negative bending moments are caused at the intermediate supports, and the bending moment diagrams have to be correspondingly revised. The continuity also affects the reactions, and therefore the shear diagrams.

In the case of equal spans (each of length l), carrying a uniformly distributed load of intensity w per unit length, the following coefficients enable the reactions and support bending moments to be ascertained.

TABLE 11.—REACTIONS FOR CONTINUOUS BEAMS OVER EQUAL SPANS.

(Coefficients of wl)

No. of spans.	Number of each support.									
	1st.	2nd.	3rd.	4th.	5th.	6th.	7th.	8th	9th.	10th
2	$\frac{3}{8}$	$\frac{10}{8}$	$\frac{3}{8}$							
3	$\frac{4}{10}$	$\frac{11}{10}$	$\frac{11}{10}$	$\frac{4}{10}$						
4	$\frac{11}{28}$	$\frac{32}{28}$	$\frac{26}{28}$	$\frac{32}{28}$	$\frac{11}{28}$					
5	$\frac{15}{38}$	$\frac{43}{38}$	$\frac{37}{38}$	$\frac{37}{38}$	$\frac{43}{38}$	$\frac{15}{38}$				
6	$\frac{41}{104}$	$\frac{118}{104}$	$\frac{100}{104}$	$\frac{106}{104}$	$\frac{100}{104}$	$\frac{118}{104}$	$\frac{41}{104}$			
7	$\frac{56}{142}$	$\frac{161}{142}$	$\frac{137}{142}$	$\frac{143}{142}$	$\frac{143}{142}$	$\frac{137}{142}$	$\frac{161}{142}$	$\frac{56}{142}$		
8	$\frac{153}{388}$	$\frac{440}{388}$	$\frac{374}{388}$	$\frac{392}{388}$	$\frac{386}{388}$	$\frac{392}{388}$	$\frac{374}{388}$	$\frac{440}{388}$	$\frac{153}{388}$	
9	$\frac{209}{530}$	$\frac{601}{530}$	$\frac{511}{530}$	$\frac{535}{530}$	$\frac{529}{530}$	$\frac{529}{530}$	$\frac{535}{530}$	$\frac{511}{530}$	$\frac{601}{530}$	$\frac{209}{530}$

TABLE 12.—SUPPORT BENDING MOMENTS FOR CONTINUOUS BEAMS OVER EQUAL SPANS.

(Coefficients of wl^2).

No. of spans.	Number of each support.									
	1st.	2nd.	3rd.	4th.	5th.	6th.	7th.	8th.	9th.	10th.
2	0	$\frac{1}{8}$	0							
3	0	$\frac{1}{10}$	$\frac{1}{10}$	0						
4	0	$\frac{3}{28}$	$\frac{2}{28}$	$\frac{3}{28}$	0					
5	0	$\frac{4}{38}$	$\frac{3}{38}$	$\frac{3}{38}$	$\frac{4}{38}$	0				
6	0	$\frac{11}{104}$	$\frac{8}{104}$	$\frac{9}{104}$	$\frac{8}{104}$	$\frac{11}{104}$	0			
7	0	$\frac{15}{142}$	$\frac{11}{142}$	$\frac{12}{142}$	$\frac{12}{142}$	$\frac{11}{142}$	$\frac{15}{142}$	0		
8	0	$\frac{41}{388}$	$\frac{30}{388}$	$\frac{33}{388}$	$\frac{32}{388}$	$\frac{32}{388}$	$\frac{30}{388}$	$\frac{41}{388}$	0	
9	0	$\frac{56}{530}$	$\frac{41}{530}$	$\frac{45}{530}$	$\frac{44}{530}$	$\frac{44}{530}$	$\frac{45}{530}$	$\frac{41}{530}$	$\frac{56}{530}$	0

LOADS ON BUILDINGS.

The following equivalent dead loads are specified in various regulations for the live loads to be taken in the design of floors, etc., of buildings.

TABLE 13

Kind of Floor.	London.		New York.	Chicago.	Philadelphia.	Boston.
	Steel Frame.	Reinforced Concrete.				
Dwelling house	70	70	60	40	70	50
Hospital wards, hotel bedrooms, etc.		84	60	40	70	50
Offices (all floors)	100	100		100	100	100
,, (first floor)	—	—	75	—	—	—
,, (other floors)	—	—	150	—	—	—
Assembly rooms, theatres, etc.	—	112	90	100	120	150
Workshops or retail shops	112	112	120	100	120	125
Ball rooms	—	150	90	100	70	200
Warehouses	224	224	150	100	150	250
Roofs Inclination > 20°	28	28	30*	25*	—	—
,, ,, not > 20°	56	56	50	25	—	40
	including wind and snow.					
Wind pressure	30 on the upper two-thirds.	20 on whole building 40 on all parts above roof.	30	20	30	30
Figures are in pounds per square foot.			* On the horizontal projection.			

TABLE 14.—CORRUGATED GALVANIZED SHEETS AND FITTINGS.

Approximate Weights in lb. per sq. ft.

	GAUGE.				
	18	20	22	24	26
Without laps	2·41	1·85	1·55	1·29	0·95
With 6-in. end lap and single side lap	2·73	2·09	1·75	1·46	1·08
With 6-in. end lap and double side lap	3·03	2·33	1·95	1·62	1·19

STRENGTH OF SHEETS.

Theoretical formula from section modulus calculated for cycloidal curve.

$$B = \frac{4fbdt}{15}$$

where B = safe B.M. in inch-tons,
f = safe stress in tons per square inch,
b = breadth in inches of sheet *before corrugating,*
d = depth in inches of corrugations,
t = thickness in inches corrugations.

∴ for uniform load on a span of l feet taking $f = 7{\cdot}5$,

$$\text{the safe load W} = \frac{4bdt}{3l} \text{ tons.}$$

Pencoyd formula (taking safe load = ¼ crippling load),

$$W = \frac{24{,}500tb'd}{l} \text{ lb.}$$

where b' = centre to centre of corrugations in inches.

TABLE 15.—WEIGHTS OF GALVANIZED FITTINGS.

Cup-headed rivets (¼″)	⅜″ long	2·0 lbs. per gross
”	½″ ”	2·15 ”
”	⅝″ ”	2·3 ”
Bolts and nuts (¼″)	¾″ ”	3·6 ”
”	1¼″ ”	4·7 ”
”	1½″ ”	5·1 ”
Hook bolts	4″ × 5/16″	20·4 ”
”	4″ × ⅜″	28·0 ”
”	4½″ × 5/16″	22·4 ”
”	4½″ × ⅜″	32·0 ”
”	5″ × 5/16″	24·9 ”
”	5″ × ⅜″	37·3 ”
Roofing screws (¼″)	2½″ long	5·3 ”
”	3″ ”	7·0 ”
Roofing pins (¼″) ..	2½″ ”	5·1 ”
” ..	3″ ”	5·9 ”
Washers (flat) ..	.. ..	1·96 ”
” (curved)	.. ..	5·22 ”

STRESSES IN PERFECT OR FIRM FRAMED STRUCTURES.

When the forces, including the reactions, acting on a perfect frame are known, the stresses in the individual members of the frame can be found by either of the following methods—

(1) Clerk-Maxwell's Reciprocal Figure method.
(2) The method of Moments or Sections, or Ritter's method.

In any important structure the stresses in all the members are obtained by one of these methods, and those in some of the members are checked by the other method.

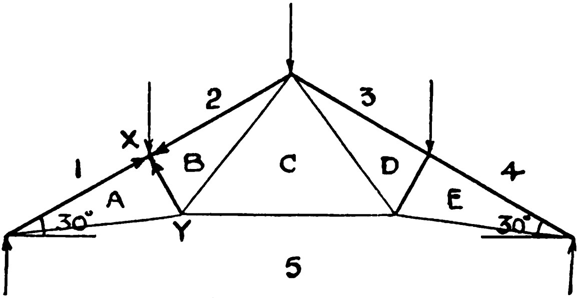
RECIPROCAL FIGURES.

Two figures consisting of lines and points lying in a plane are said to be reciprocal when—

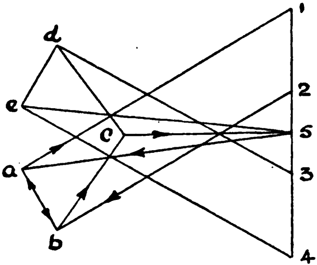
(1) To any *node* or point of one figure at which a given number of lines meet, there is a corresponding *polygon* in the other figure, bounded by the same number of sides.

(2) To every line of one figure there corresponds a parallel line in the other figure.

(3) To a line of one figure joining two nodes there corresponds a line in the other figure separating the polygons corresponding to these nodes.



Frame Diagram.



Reciprocal Diagram.

FIG. 3.—STRESSES IN SIMPLE ROOF TRUSS.

Clerk-Maxwell enunciated the theorem that if one of these figures represents a framework with the forces acting on it, the other or reciprocal figure will give the forces on the framework and the stresses in the individual members.

We see, therefore, that we can find graphically the stresses in a framework by drawing its reciprocal figure.

Example of Simple Roof Truss. Take the case of the simple roof truss shown. In this method we will adopt Bow's notation of lettering or numbering the spaces between the bars or forces. In this case we take the vertical loads on the nodes as equal, the reactions then being equal and vertical.

To commence the reciprocal figure set down lengths 1, 2 ; 2, 3, etc., on a vertical line to represent the forces, to some convenient scale, the reaction 4, 5 being equal to half the total load, and giving the point 5 as shown. At the left-hand end of the truss three lines meet, viz., 5, 1 ; 1, A ; A, 5. On the reciprocal figure, therefore, we require a corresponding triangle, so draw 1, *a* parallel to 1, A, and 5, *a* parallel to 5, A, their point of intersection determining the point *a* on the reciprocal figure. From *a* draw *a b* parallel to A B, and 2 *b* parallel to 2 B, thus obtaining the point *b* ; then *b c* parallel to B C, and 5 *c* parallel to 5 C, thus obtaining the point C, and so on.

To serve as a check on the accuracy of the drawing, the line joining the last point *e* on the reciprocal figure to the point 5 should be parallel to the bar E 5 of the frame.

Then the lengths of the lines of the reciprocal figure give to the scale to which the loads were set down—the stresses in the corresponding bars of the frame.

To Distinguish Between Ties and Struts. To ascertain which members of a framework are ties and which are struts, the following method is adopted and can be applied for all systems of loading.

Consider any one of the nodes of the truss at which the direction of one force is known, say the node X. Corresponding to this node we have the polygon 1 2 *b a* 1 on the reciprocal figure. The direction of the force 1 2 is known to be vertically downward, so continue the arrow-heads in this direction round the polygon 1 2 *b a* 1. Now transfer the direction of these arrow-heads to the corresponding bars close to the given node. Then if the arrow-head on a given bar points towards the node, the bar is a strut ; and if it points away, the bar is a tie. In this way it is seen that the bars 1A, AB, and B2 are all struts.

Now consider the node Y. Corresponding to this we have the polygon 5 *a b c* 5. Since AB is a strut, the arrow-head at the node Y points towards the node, and so the arrow heads go round the polygon in the direction *a b*, *b c*, *c* 5, 5 *a*, as shown. Transferring these arrow-heads to the Frame Diagram, we see that the bars B C, C 5, and 5 A are all ties.

With practice one can tell by inspection in most cases whether a given bar is a strut or a tie by the following rule. If, on imagining the given bar cut through, the forces would tend to increase its length such bar is a tie ; if the forces tend to decrease its length, the bar is a strut.

Example of Warren Girder with Uneven Loading. As a further example of reciprocal figures take the example of the Warren girder loaded as shown.

We must first find the reactions before we can proceed with the reciprocal figure. These reactions, found in the ordinary manner by moments, come 4·75 and 5·25 tons at the left and right hand ends respectively. Choosing a suitable force scale, we set down 1,2 and 2, 3 to represent 2 and 5 tons respectively; next set up 3,4 to represent the reaction of 5·25 tons; and then set down 4,5; 5,6; and 6,7 to represent 1 ton each, the length 7,1 checking back to give the reaction 4·75 tons. We now proceed as before drawing 1*a* parallel to 1A and 7*a* parallel to 7A; then *ab* and 1*b* parallel respectively to AB and 1B, and so on, the reciprocal figure coming as shown, and *l*3 coming parallel to L3, and thus serving as a check on the drawing.

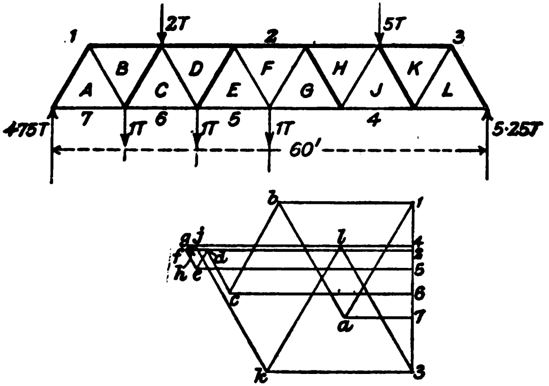


FIG. 4.—STRESSES IN WARREN GIRDER.

Wind on Roof with Curved Rafter. This case is more complicated than the previous ones because the wind pressure will have different intensities according to the slope, and the forces on the frame are at different inclinations. The manner in which such a case is worked will be clear from considering the case shown. This roof truss is of 50 ft. span and 8 ft. rise, the nodes lying on arcs of circles, the depth of the truss being 8 ft. The end X is fixed and the end Y is free, and the wind is blowing on the fixed end. The inclination of the bays 2A, 3B, 4D are measured and found to be 55°, 37°, and 19° respectively. Using a vertical pressure of 56 lbs. per sq. ft. the pressures on the bars —obtained from tables or curves—are 49, 39, and 23 lbs. per sq. ft. respectively. Multiplying these by area covered by each bay, i.e. length of bay × distance between principals, the forces carried by each bay are 2, 1·58 and ·94 tons respectively. Placing half of these at the ends of the bays we get the wind forces on the truss. Now set these forces down on a vector line, i.e. 12 = 22′ = 1 ton ; 2′3 = 33′ = ·79 ton ; 3′4 = 45 = ·47 tons ; then joining 2, 3 ; 3, 4 we get the

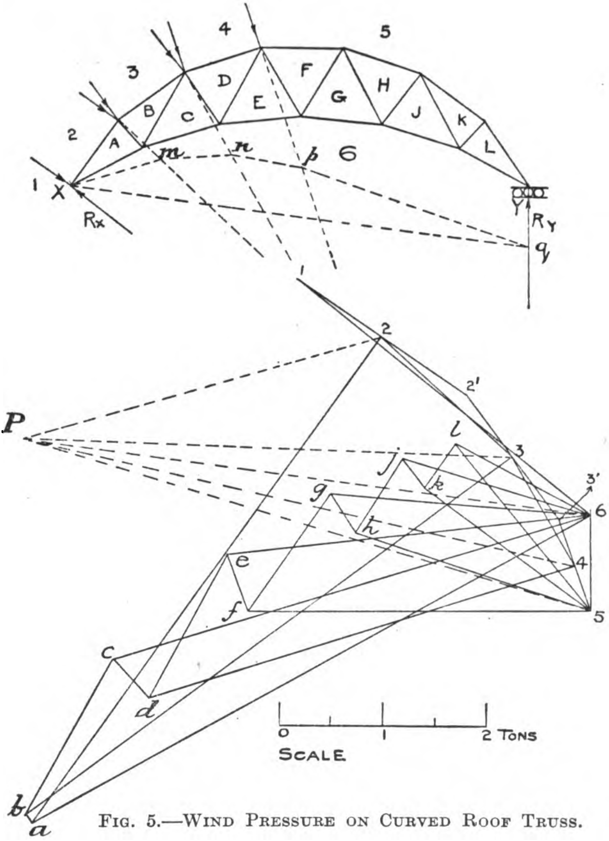


FIG. 5.—WIND PRESSURE ON CURVED ROOF TRUSS.

resultant forces 2, 3 ; 3, 4. To get the values of the reactions we draw parallels through the nodes to the resultant forces at them and take any pole P ; then as before we start at X and draw the link polygon X m, n, p, q, the first link being parallel to P2, and draw a parallel through P to Xq to cut the vertical through 5 in 6, then 56 = R_Y, 61 = R_X. Having obtained the point 6 the stress diagram can be drawn without much trouble, but great care must be taken in seeing that long lines such as 2a are accurately parallel to their comparatively short bars. In some cases on important work it is advisable to calculate the inclination of such bars in order to get accurate parallels to them.

THE METHOD OF MOMENTS OR SECTIONS.

This method is also known as Ritter's method, and in some cases, such as parallel flange girders, it is just as quick as the reciprocal figure method, and in other cases it is very useful as a check on the stresses in some of the bars found by the latter method. Let ABCD represent one bay of a framed structure. Suppose we cut it by a line XX. Then since, if the actual bars were cut the whole structure would collapse, it follows that the forces in the bars AB, BD, DC must neutralise the forces acting on the structure to the right or left of the line XX; therefore, the moment of the forces in these bars about any point whatever must be equal to the moment about that point of all the external forces to the right or left of XX. Thus, by taking moments about one of the points where two bars meet, we see, since the moment of a force about a point in its line of action is zero, that the moment of the force in the remaining bar about such point is equal to the moment of the external forces about that point. If, therefore, we require the force

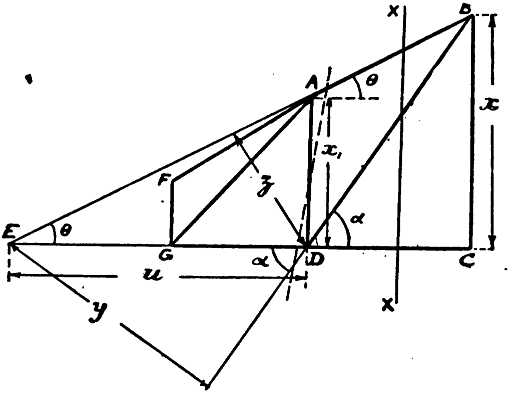


FIG. 6.—THE METHOD OF MOMENTS.

in AB, take moments about the point D, then moment of force in AB about D

$$
\begin{aligned}
&= f_{AB} \times z = \text{moment of external forces about D}\\
&\qquad\quad = \text{bending moment at D}\\
&\qquad\quad = M_D
\end{aligned}
$$

Similarly, to get force in CD take moments about B, then

$$f_{DC} \times x = M_B$$

Again, to get force in BD, take moments about E, where BA and CD produced, meet, then

$f_{BD} \times y$ = moment of external forces to left or right of XX about E.

Vertical Members in Method of Sections. If it is required to find the stress in a vertical member of a framed structure by means of the method of sections, we have only to imagine it slightly inclined. If, for example, the stress in AD is required, imagine it slightly inclined, then two of the bars DG and BA meet in E.

$\therefore f_{AD} \times u$ = moment about E of forces to right or left of given section.

Method of Sections applied to Graphical Construction. Let AKG be a truss or other framed structure loaded in any manner, and let the BM diagram, drawn with a polar distance p for the given loading treated as loads on a simple beam be *abcdefg*. Then, as previously explained, considering one bay,

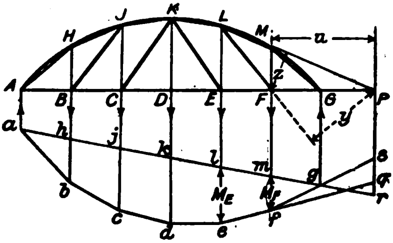


FIG. 7.—METHOD OF MOMENTS USING BM DIAGRAMS.

$$f_{LM} = \frac{M_F}{z} = \frac{p \times mf}{z}$$

$$f_{EF} = \frac{M_E}{x} = \frac{p \times le}{x}$$

$$f_{LF} = \frac{\text{moment of forces to left of E about P}}{y}$$

$$= \frac{p \times qr}{y}$$

Where q and r are points on vertical through P, where lm and ef produced cut it. This follows from the reasoning for the link and vector polygon construction given on p. 172.

Similarly—

$$f_{MF} = \frac{\text{moment of forces to left of F about P}}{u}$$

$$= \frac{p \times qr}{u}$$

Where s is point where fg produced cuts vertical through P.

Girders with Parallel Flanges. The method of sections is particularly simple in the case of girders with parallel flanges, because the depth in each case is constant.

Take a Linville truss loaded in any way.

Then stress in CD $= f_{CD} = \frac{M_D}{d}$; stress in PO $= f_{PO} = \frac{M_C}{d}$

Therefore to some scale the B.M. diagram gives the stresses direct, or by drawing the B.M. diagram with a polar distance equal to the depth d, the B.M. diagram will give the stresses to the load scale.

When we come to obtain the stresses in the verticals and diagonals, we cannot take moments as before because the top and bottom flanges do not meet at finite distances.

Consider any diagonal, say OC; since PO and CD are horizontal, the forces in them cannot have any effect on the vertical force acting on the bay. Therefore, vertical component of force in OC = resultant vertical force over bay = what we have previously called the *shearing force* on the bay. We thus see that we obtain the stresses in the diagonals by resolving the shearing force in the direction of the diagonal, i.e. drawing cd parallel to OC we get the forces in OC. Similarly, the force in any vertical is equal to the shearing force at the given point, i.e. force in CP = ce.

Girder with Parallel Flanges—Trigonometrical Solution. In the case shown in the diagram on page 168, if all the diagonals are of the same angle the stresses can be very simply calculated as follows—

Take first the stresses in flanges.

$$f_{AB} = \frac{R_A \cdot AB}{d} = R_A \cot \theta$$

$$f_{QP} = \frac{R_A \cdot AB}{d} = R_A \cot \theta$$

$$f_{BC} = \frac{R_A \cdot AC - W_1 \, BC}{d} = (2 R_A - W_1) \cot \theta = f_{PO}$$

$$f_{CD} = \frac{R_A \cdot AD - W_1 BD - W_2 CD}{d} = (3R_A - 2W_1 - W_2) \cot \theta = f_{DN}$$

and so on.

If we take the *nth* bay from end A, the diagonals remaining in the same direction,
Stress in lower flange

$$= [n\ R_A - (n - 1)\ W_1 - \ n - 2)\ W_2 . \ldots - W_{n-1}]\ \cot\theta$$

Stress in upper flange

$$= [(n - 1.)\ R_A - \ (n - 2)\ \ W_1 \ldots . W_1]\ \cot\theta$$

Now take the diagonals.

$$f_{AQ} = R_A \operatorname{cosec}\theta$$

$f_{BP} = R_A \operatorname{cosec}\theta - W_1 \operatorname{cosec}\theta - R_A = W_1\ (R_A - W_1) \operatorname{cosec}\theta$ and so on.

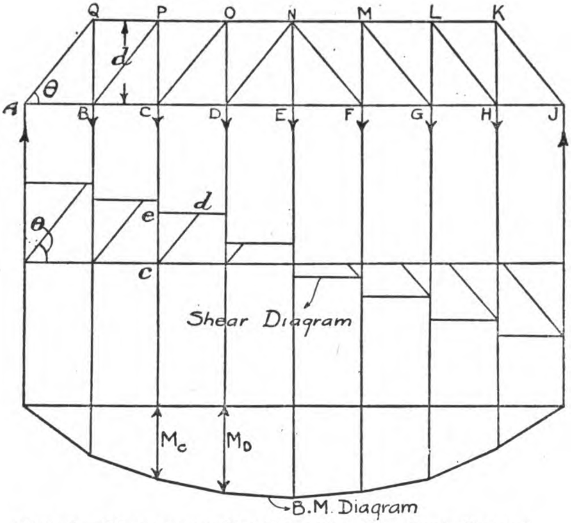


FIG. 8.—METHOD OF MOMENTS FOR PARALLEL GIRDER.

Stress in *nth* bay, the diagonals remaining in constant direction,

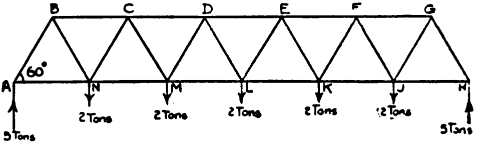
$$f^{\text{nth}}_{\text{diagonal}} = \operatorname{cosec}\theta .\ (R_A - W_1 \ldots . - W_{n-1})$$

Finally take the verticals,

$$f_{QB} = R_A - W_1$$
$$f_{PC} = R_A - W_1 - W_2$$

and so on.

$$f^{\text{nth}}_{\text{vertical}} = R_A - W_1 - \ldots . \ W_n$$



Example. Take the Warren girder shown above. For $\theta = 60°$, $\cot\theta = \cdot577$. $\operatorname{cosec}\theta = 1\cdot155$.

$\therefore f_{AN} = 5 \cot 60° = 5 \times \cdot577 = 2\cdot885$ tons $= f_{HJ}$

$f_{NM} = (5 \times 3 - 2) \cot 60° = 13 \times \cdot577 = 7\cdot501$ tons $= f_{JK}$

$f_{ML} = (5 \times 5 - 2 \times 3 - 2 \cot 60° = 17 \times \cdot577 = 9\cdot909$ tons $= f_{KL}$

$f_{BC} = (5 \times 2) \cot 60° = 5\cdot770$ tons $= f_{GF}$

$f_{CD} = (5 \times 4 - 2 \times 2) \cot 60° = 16 \times \cdot577 = 9\cdot232$ tons $= f_{FE}$

$f_{DE} = (5 \times 6 - 2 \times 4 - 2 \times 2) \cot 60° = 18 \times \cdot577 = 10\cdot386$ tons $— f_{ED}$

$f_{AB} = f_{BN} = 5 \operatorname{cosec} 60° = 5 \times 1\cdot155 = 5\cdot775$ tons $= f_{HG} = f_{GJ}$

$f_{NC} = f_{CM} = (5 - 2) \operatorname{cosec} 60° = 3 \times 1\cdot155 = 3\cdot465$ tons $= f_{JF} = f_{FK}$

$f_{M} = f_{DL} = (5 - 2 - 2) \operatorname{cosec} 60° = 1\cdot155 = 1\cdot155$ tons $= f_{KE} = f_{EL}$

Stresses in Tripods and Shear Legs. Although the general treatment of space frames is beyond our present scope, we will deal with the stresses in shear legs or tripods.

Draw the structure in plan and elevation, and let W be the load at A, AB being the back leg and AD, AE the fore legs. Resolve W down AB and *down the plane* of the other two legs, i.e. set out *ab* equal to W and draw *bc* parallel to AB, and *ac* parallel to AC, then *bc* is the force in AB. Now swing the shear legs down horizontally in order to get A_2DE, the true shape of the triangle ADE, then setting out *ac* horizontally and drawing *ad* and *cd* parallel to EA_2, DA_2 respectively, we get the stresses in the fore legs.

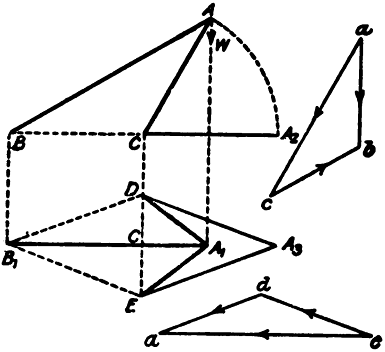


FIG. 9.—STRESSES IN SHEAR LEGS.

MECHANICS.

Force is that which tends to produce or modify motion, and it is expressed in units of pounds or tons.

It is important to remember that for every Force or Action, there is a Reaction, and that Action and Reaction are equal and opposite.

If a number of forces act on a body, each one has its effect which is independent of the others.

Work is said to be done whenever a body is moved by a force, and is measured by the product of the force employed by the distance through which the body on which it acts is moved in the direction of the force. The unit is the foot-pound. Sometimes the foot-tons and similar convenient units are used.

Power is the work done in a given time, the principal unit being foot-pounds per minute.

Horse-power is the engineering unit, which is equal to 33,000 foot-pounds per minute.

FORCE.

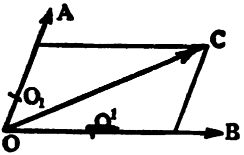
A force has magnitude and is applied along a line in a certain definite direction. A force can, therefore, be represented by a line, for a line may be drawn of any magnitude, through any point, and along any direction; and an arrow will indicate the "sense" in which the force is applied, that is, whether it is toward or away from a given point.

A single force which can replace two forces is known as the Resultant, and the separate forces are known as the Components.

Convenient graphical methods of finding the Resultant and of solving similar problems are given on pages 170, 171, 172, and 173.

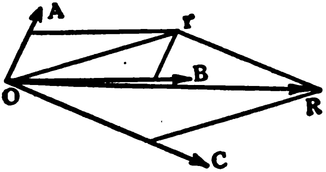
PARALLELOGRAM OF FORCES.

Two forces are represented by the lines OA, OB. To find the *Resultant*, complete the parallelogram OACB. Then OC represents the Resultant in Magnitude Direction and Sense. If OA is $\frac{11}{32}''$ and represents a pull of 11 lbs., if OB is $\frac{21}{32}''$ and represents a pull of 21 lbs., then using the scale of $\frac{1}{32}''$ per 1 lb., the resultant will be a pull of 27 lbs., for OC is $\frac{27}{32}''$. If the forces are not given as acting through the same point O but at O1 and O1 respectively, the lines O1A, and O1B should be produced to intersect at O, and the parallelogram constructed as before.



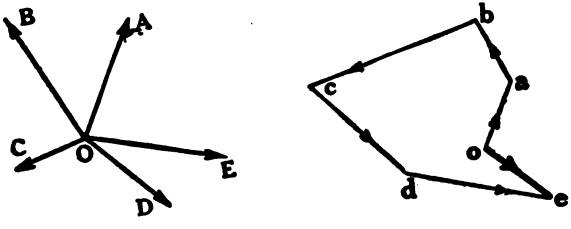
RESULTANT OF THREE FORCES.

Three forces OA, OB, OC act at a point. Construct the parallelogram for OA, OB, the resultant is O*r*. Replacing OA and OB by O*r*, we have to find the resultant of O*r*, OC. Draw the parallelogram therefor, and the resultant OR is the resultant of the three forces OA, OB, OC.



POLYGON OF FORCES.

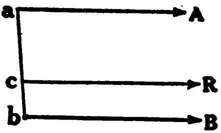
If in the preceding problem, instead of drawing two parallelograms, we had started with OA, drawn A*r* equal and parallel to OB, and then *r*R equal and parallel to OC, we should get the resultant OR right off. When any number of forces OA, OB OC, OD, DE meet at a point, the resultant is obtained by drawing *oa* parallel to and proportional to OA; *ab* parallel and proportional to OB, and so on. By joining starting-point *o* and the last point *e*, we get the resultant as proportional to *oe* and with a direction parallel to it. If *oe* can replace the forces OA, OB, etc., then *eo*, equal and opposite thereto, can balance them, and this force is known as the equilibriant.

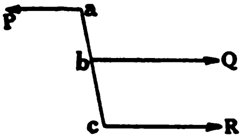


PARALLEL FORCES.

Parallel Forces A, B, acting in the same sense at points *ab*. The resultant R = A + B, and is parallel to A and B. Join *a* and *b*, and divide it inversely as the forces so that *ac* : *cb* : : B : A.

Then *c* is the position at which R should be applied.





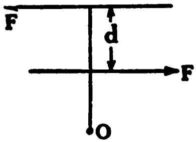
When parallel forces P and Q act in opposite senses, the resultant R = Q − P.

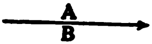
Produce *ab* to *c* such that *ac* : *cb* : : Q : P.

Moment of a Force.

The moment of a force F with respect to a point O is obtained by multiplying the force F by *a*, the perpendicular from O to the line of action of the Force. This moment is usually called the first moment to distinguish it from the second moment to be dealt with later.

If the distance is measured in feet and the force in pounds, the moment is expressed in foot-pound units.





The sum of the moments of any number of forces acting in the same plane with respect to any given point is equal to the moment of their resultant.

If two oppositely-acting parallel forces F are considered, it is seen that they have no resultant. They are called a Couple and tend only to produce rotation with a moment of constant value, viz., F × *d*. This value is independent of the position of the turning point O.

A couple can only be balanced by a couple.

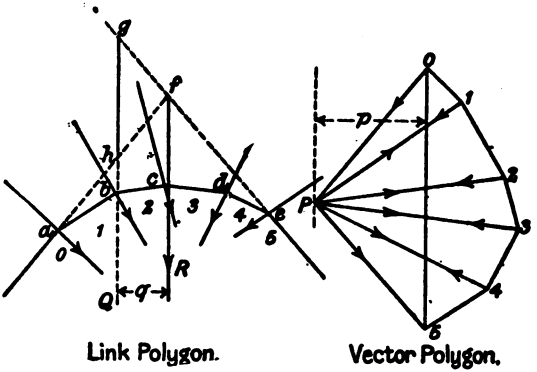
Bow's Notation. One method of distinguishing a force is to employ two letters such as AB, one on each side of the line of action.

GRAPHIC STATICS.

Link and Vector Polygon Construction.

Suppose that 1, 2 ; 2, 3 ; 3, 4 represent a number of forces in one plane. To some convenient scale, draw 0, 1 on a vector figure to represent the force 0, 1 in magnitude and direction; then from 1 draw 1, 2 to represent the force 1, 2 in magnitude and direction, and so on until the last force 4, 5 is reached. Then the line joining the first point in the vector figure to the last point, i.e. 0, 5, will give the magnitude and direction of the resultant of the forces, and the line 0, 5 is called the closing line of the *vector polygon*. Now if the given forces are in equilibrium they can of course have no resultant, and so the first and last points of the vector polygon of a number of forces in equilibrium must coincide.

In dealing with problems in which we wish to find the resultant of a number of forces, we usually require to know the *position* as well as the magnitude and direction of the resultant; and



Link Polygon. Vector Polygon.

FIG. 10.—LINK AND VECTOR POLYGON CONSTRUCTION.

unless all the forces pass through the same point, in which case the resultant will also pass through that point, some separate construction must be used. Such construction is known as the *link and vector polygon* construction, and is as follows. Let 0, 1; 1, 2 (Fig. 10), and so on, be a number of forces not necessarily parallel nor concurrent. To some suitable scale set down on a vector figure 0, 1, 2, and so on, then as before the closing line 0, 5 gives the magnitude and direction of the resultant. Now take any point or pole P at any convenient position on the paper and join P, 0; P, 1; and so on. Then draw anywhere across the line of action of the first force a line *a, f*, parallel to P, 0 and cutting the line of action of the force in *a*; across space 1 draw *a, b* parallel to P, 1. across space 2 *c, d* parallel to P, 3; and so on until the last line or link parallel to P, 5 is reached. Produce this last link to meet the first link in *q*, then the resultant R will pass through the point *q*, and the figure *abcdef* is called the *link polygon* or the funicular polygon.

GRAPHICAL DETERMINATION OF THE FIRST MOMENT OF A NUMBER OF FORCES ABOUT A GIVEN POINT.

This we obtain by means of the link and vector polygon construction (see Fig. 10).

Suppose the moment of the given force system is required about the point Q. Through Q draw a line parallel to the resultant R, to cut the first and last links produced in *h* and *g*. Then if the point P is at perpendicular or polar distance *p* from 0, 5 on the vector figure, moment of force system about Q is equal to *gh* × *p*, *gh* being read on the space scale and *p* on the force scale.

GRAPHICAL INTEGRATION OR SUM CURVE CONSTRUCTION.

If a curve is drawn on a horizontal base and a new curve be drawn, such that its ordinate at any point represents the area of the given curve up to that point, the new curve is called the *sum curve* or *integral curve* of the given curve, which is called the *primitive curve.*

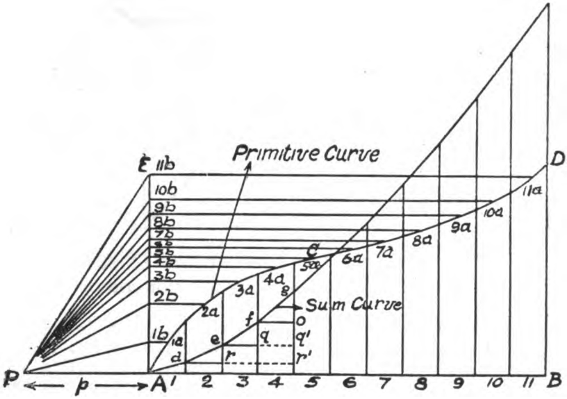


FIG. 11.—SUM CURVE CONSTRUCTION.

The sum curve can be obtained graphically as follows. Let ACD be any primitive curve on a straight base AB. Divide AB into any number of parts, not necessarily equal (but for convenience of working they are generally taken as equal). These so-called base elements should be taken so small that the portion of the curve above them may be taken as a straight line. About 1 cm. or ·4 ins. will usually be a suitable size and in most cases a smaller element 11 will come at the end. Find the mid-points, 1, 2, 3, etc., of each of the base elements and let the verticals through these mid-points meet the curve in 1*a*, 2*a*, 3*a*, etc. Now project the points on to a vertical line AE, thus obtaining the points 1*b*, 2*b*, 3*b*, etc., and join such points to a pole P on AB produced and at some convenient distance *p* from A. Across space 1 then draw A*d* parallel to P 1*b*; *de* across space 2 parallel to P 2*b*, and so on, until the point *n* is reached. Then the curve A*de* . . . . *n* is the sum curve of the given curve, and to some scale B*n* represents the area of the whole curve.

To obtain the scale of these ordinates we multiply the vertical scale by the length *p* measured on the horizontal scale.

FIRST MOMENT OF AN AREA.

Let a small element of area *a* of any figure be situated at the point P, and let VV be any straight line or axis. Then if PN is drawn perpendicular to VV, $a \times$ PN is the first moment of the element of area about the given line. Now, if the whole figure is divided up into elements of area such as a, and the moments of each element be taken about VV, and the whole of these moments be added together, the resulting sum is called the *first moment of the area.*

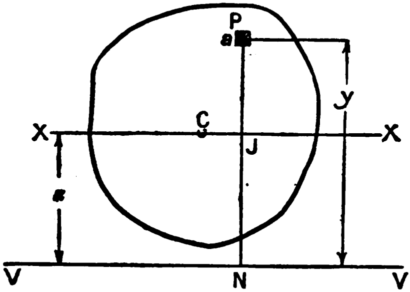


FIG. 12.—MOMENT OF AREA.

∴ The first moment of the whole area is the sum of quantities such as $a \times$ PN. This is expressed symbolically as follows—

First moment of whole area $= \Sigma\ (a \times \text{PN})$.

Now the *centroid* or *the first moment centre* of an area is defined as the point at which the whole area can be considered concentrated, in order that its moment about any given line will be equal to the first moment of the area about the same line.

Thus if C is the centroid of the area, and lies on a line XX drawn parallel to VV at a distance x from it, and the area of the whole figure is A, we have—

$$\text{A} \times x = \Sigma\ (a\ .\ \text{PN})$$

$$\therefore\ x = \frac{\Sigma\ (a\ .\ \text{PN})}{\text{A}}$$

This will not determine the exact position of C, but only its distance from VV; to obtain its exact position we must also find its distance from some other line not parallel to VV.

The centroid will lie on every *line of symmetry* of the area, i.e. each line which divides the area into two exactly similar right and left-hand portions.

The first moment of any area is zero about any line passing through its centroid.

MOMENT OF INERTIA AND RADIUS OF GYRATION OF SECTIONS.

If any section of area A be considered as divided up into a very large number of elements a such as that around the point P, and a perpendicular be drawn to any line VV, then the moment of inertia (or second moment) of this element of area about the line VV is given by $a \,.\, PN = ay^2$, and by adding together the moments of inertia of all the elements of area about this line we obtain the total moment of inertia of the whole section about the line. This is written

$$I_V = \Sigma ay^2. \qquad (1)$$

MOMENT OF INERTIA ABOUT A PARALLEL LINE THROUGH THE CENTROID.

If C is the centroid of the area and XX is drawn through C parallel to VV and at a perpendicular distance x from it we have if A is the area of the section

$$I_X = I_V - Ax^2. \qquad (2)$$

RADIUS OF GYRATION (g).

This is defined by the relation

$$Ag^2 = I.$$

$$\therefore g = \sqrt{\frac{I}{A}} = \sqrt{\frac{\text{Moment of Inertia}}{\text{Area}}}; \qquad (3)$$

$$\text{i.e. } g_V = \sqrt{\frac{I_V}{A}}$$

$$g_X = \sqrt{\frac{I_X}{A}}$$

$$\therefore \text{In equation (2) } Ag_X^2 = Ag_V^2 - Ax^2$$

$$\text{or } g_X^2 = g_V^2 - x^2 \qquad (4)$$

CALCULATIONS FOR BUILT-UP SECTIONS.

Tables of values of the moments of inertia of a large number of sections are given on pages 183 to 188. To compute the moment of inertia or radius of gyration of a section built up of these sections we add together the moments of inertia of the separate portions, using formula (2) to transfer the lines about which the moments are taken. To obtain the radius of gyration apply formula (3) to the moment of inertia and area of the whole section.

The radius of gyration of a compound section cannot be obtained by adding together the radii of gyration of the separate parts.

NUMERICAL EXAMPLE.

Take the built-up girder section shown.

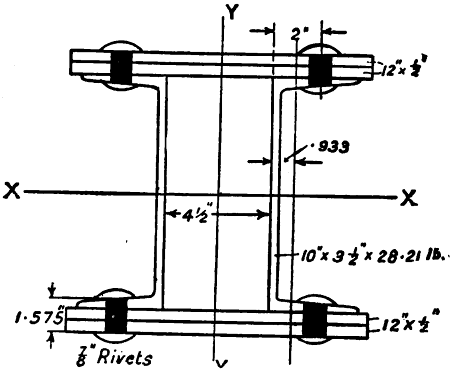


FIG. 13.—INERTIA OF BUILT-UP GIRDER SECTION.

MOMENT OF INERTIA ABOUT XX (neglecting rivet holes).

I_X of two channels (p. 184) $= 2 \times 117{\cdot}9$ 235·8

21_X of 2 – 12 ins. × 1 in plates 10 ins. apart

$$= 12\left(\frac{12^3-10^3}{12}\right) = 12 \times 60{\cdot}67 \quad 728{\cdot}0$$

Total I_X 963·8

Deduction for Rivets.

$$\text{I of rivets} = 2 \times \tfrac{7}{8}\left(\frac{12^3 - 8{\cdot}85^3}{12}\right) \qquad 150{\cdot}8 \text{ ins.}$$

Net I_X $813{\cdot}0$ ins.^{4}

Approximate allowance for rivets: regard them as 1½ ins. thick 9 ins. apart $= 2 \times \frac{7}{8} \times 83{\cdot}2 = 146$.

This would give $I_X = 818$ which is sufficiently close for most engineering calculations.

RADIUS OF GYRATION ABOUT XX.

$$A = 24 + 2 \times 8{\cdot}296 = 40{\cdot}59.$$

$$\therefore g_X = \sqrt{\frac{963{\cdot}8}{40{\cdot}59}} = 4{\cdot}87 \text{ ins.}$$

Allowance for Rivets.

$$A = 40{\cdot}59 - 4 \times \tfrac{7}{8} \times 1{\cdot}575 = 35{\cdot}08.$$

$$\therefore g_X = \sqrt{\frac{813{\cdot}0}{35{\cdot}08}} = 4{\cdot}81 \text{ ins.}$$

The effect upon the radius of gyration of neglecting the rivet holes is practically negligible.

MOMENT OF INERTIA ABOUT YY (neglecting rivets).

I_Y of 4 plates $12 \times \frac{1}{2} = 2 \times \frac{12^3}{12}$ = 288

I_Y of 2 channels about own centroids (p. 184)

$= 2 \times 8{\cdot}2 =$ 16·4

Ad^2 for 2 channels $= 2 \times 8{\cdot}296 \times (2{\cdot}25 + {\cdot}938)^2 =$ 168·5

Total I_Y 472·9

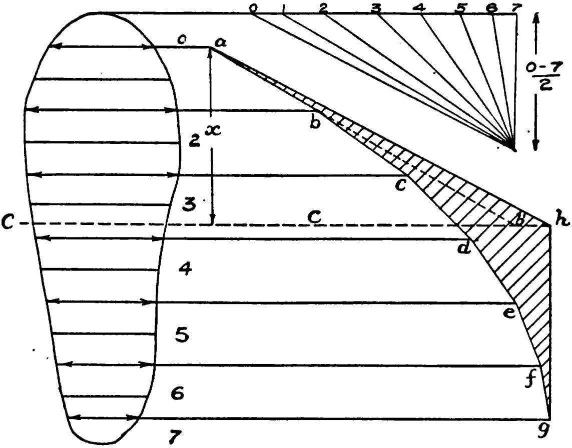


FIG. 14.—MOHR'S CONSTRUCTION FOR MOMENT OF INERTIA.

Allowance for Rivets. The rivets amount to a pair of rectangles $\frac{7}{8}$ in. thick and $2 \times 1{\cdot}575 = 3{\cdot}15$ ins. wide $7\frac{5}{8}$ ins. apart; we have for $7\frac{1}{2}$ ins. apart 30·80 per inch width and 34·57 for 8 ins.—say 31·7 for $7\frac{5}{8}$ ins.

$$\therefore \text{deduction} = 31{\cdot}7 \times 3{\cdot}15 = 100 \text{ very nearly.}$$

$$\therefore \text{nett } I_Y = \underline{372{\cdot}9} \text{ ins.}^4$$

$$g_y = \sqrt{\frac{472{\cdot}9}{35{\cdot}08}} = \underline{3{\cdot}41} \text{ ins.}$$

GRAPHICAL CONSTRUCTION FOR CENTROID AND MOMENT OF INERTIA.

The following graphical construction—known as Mohr's construction—is useful for finding the centroid and moment of inertia about the centroid of irregular figures. Divide the area up into a number of small strips, of equal breadth, parallel to the direction about which moments are taken, and draw the centre line of each of said strips. Then if the strips are sufficiently small (we have only taken a few strips in the figure to avoid complication) the lengths of these centre lines represent the areas of the separate strips. Now, on a vector line, to some scale, set out 0, 1, 1, 2, . . . 6, 7 to represent the area of each strip, and take a pole P at distance = $\frac{1}{2}$ total area 0, 7 from this vector line. Then anywhere across space 0 draw and produce a line *ah* parallel to 0, P: across space 1 draw *ab* parallel to P 1; across space 2, *bc* parallel to P 2, and so on until the point *g* is reached. Then draw the last link *gh* parallel to the last line P 7 to meet *ah* in *h*.

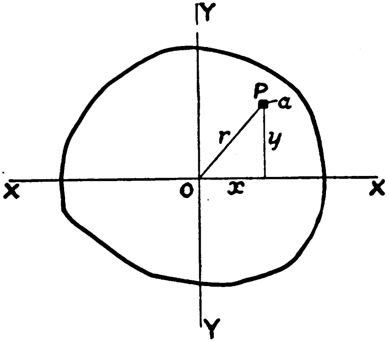


FIG. 15.—POLAR MOMENT OF INERTIA.

Then the line XX through the centroid passes through *h*, and if *a* is the area of the shaded area, and A is the area of the figure,

$$I_{CC} \text{ of figure} = A \times a.$$

POLAR MOMENT OF INERTIA.

The polar moment of an area about a pole or axis O is defined as $\Sigma a.\ r^2$, and is usually given the letter I_P. It is useful in calculations upon the energy of fly wheels.

If XX and YY are any two axes through O at right angles to each other we have

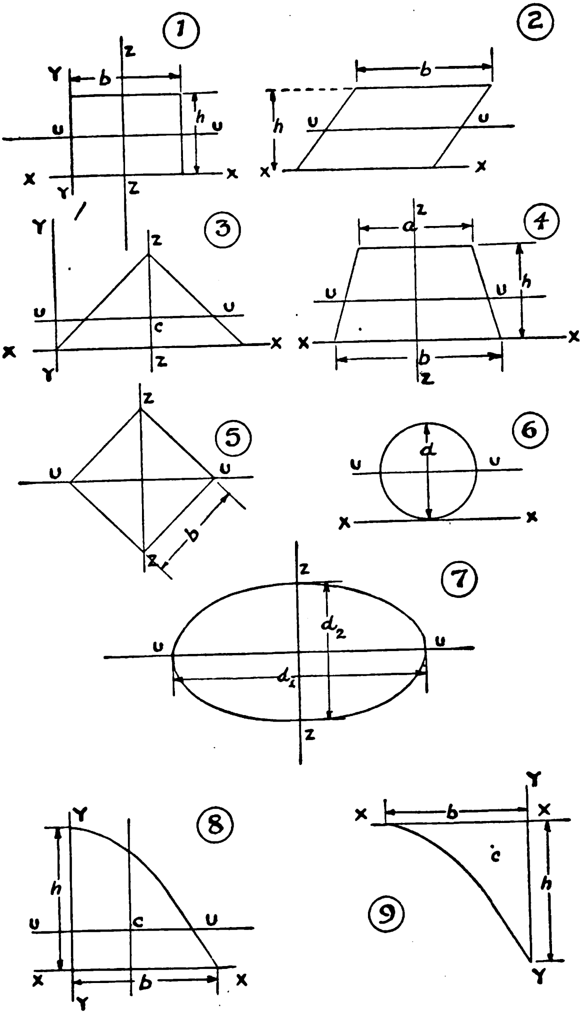
$$I_P = I_X + I_Y$$

this is because $r^2 = x^2 + y^2$

$$\therefore\ \Sigma ar^2 = \Sigma ax^2 + \Sigma ay^2$$
$$\therefore\ Ag_P{}^2 = Ag_X{}^2 + Ag_Y{}^2$$
$$\text{or } g_P = \sqrt{g_X{}^2 + g_Y{}^2}$$

TABLE 16.—AREA, POSITION OF CENTROID, AND MOMENT OF INERTIA OF COMMON FIGURES.

No.	Figure.	Area.	Position of Centroid.		Moment of Inertia.			
			From XX.	From YY.	I_{XX}.	I_{YY}.	I_{YY}.	I_{ZZ}.
1	Rectangle	bh	$\frac{h}{2}$	$\frac{b}{2}$	$\frac{bh^3}{3}$	$\frac{hb^3}{3}$	$\frac{bh^3}{12}$	$\frac{hb^3}{12}$
2	Parallelogram	bh	$\frac{h}{2}$	—	$\frac{bh^3}{3}$	—	$\frac{bh^3}{12}$	—
3	Triangle	$\frac{bh}{2}$	$\frac{h}{3}$	$\frac{b}{2}$	$\frac{bh^3}{12}$	$\frac{7hb^3}{48}$	$\frac{bh^3}{36}$	$\frac{hb^3}{48}$
4	Trapezium (where $a = n\ b$)	$\frac{hb}{2}(1+n)$	$\frac{h}{3}\left(\frac{2n+1}{n+1}\right)$	—	$\frac{bh^3}{12}(3n+1)$	—	$\frac{bh^3}{36}\frac{(n^2+4n+1)}{(n+1)}$	$\frac{bh^3}{2}(n+3)$
5	Square	b^2	—	—	—	—	$\frac{b^4}{12}$	$\frac{b^4}{12}$
6	Circle	$\frac{\pi d^2}{4}$	$\frac{d}{2}$	—	$\frac{5\pi d^4}{64}$	—	$\frac{\pi d^4}{64}$	—
7	Ellipse	$\frac{\pi d_1 d_2}{4}$	—	—	—	—	$\frac{\pi d_2^3 d_1}{64}$	$\frac{\pi d_1^3 d_2}{64}$
8	Parabolic segment (interior)	$\frac{2}{3}\ bh$	$\frac{2}{5}\ h$	$\frac{3}{8}\ b$	$\frac{16}{105}\ bh^3$	$\frac{2}{15}\ hb^3$	$\frac{8}{175}\ bh^3$	—
9	Parabolic segment (exterior)	$\frac{1}{3}\ bh$	$\frac{3}{10}\ h$	$\frac{b}{4}$	—	—	—	—



Properties of Common Figures.

Application to Rotating Bodies.

So far we have considered only the moment of inertia and radius of gyration of areas.

If we are concerned with rotating bodies, the mass of the body comes into consideration.

We then have

$$\text{Moment of Inertia} = I = \frac{Wg^2}{32{\cdot}2}$$

where g = radius of gyration in feet, of section of body about the axis of rotation,

$32{\cdot}2$ = gravity acceleration in feet per sec.^{2}

ROLLED STEEL BEAMS, ETC.

Properties of British Standard Sections.

The Tables on the following pages give the properties of the British Standard Sections which are usually listed by makers.

For the properties of British Standard Steel Sections, see p. 183

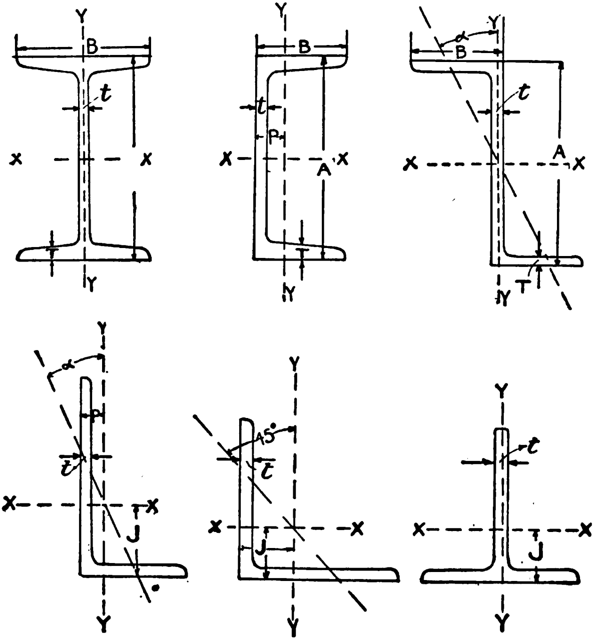


FIG. 16.—PROPERTIES OF BRITISH STANDARD SECTIONS.

BRITISH STANDARD SECTIONS.* (See Fig. 16).

TABLE 17.—PROPERTIES OF BRITISH STANDARD I BEAMS.

Size. H. B.	Wt. per Foot.	Thickness.		Sectional Area.	Moments of Inertia.		Section Moduli.		Radii of Gyration.	
		t.	T.		XX.	YY.	XX.	YY.	XX.	YY.
Inches.	Lbs.	Ins.	Ins.	Sq.ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
3 × 1½	4	·16	·248	1·18	1·66	·124	1·11	·165	1·19	·325
3 × 3	7½	·20	·332	2·50	3·89	1·16	2·53	·841	1·23	·710
4 × 1¾	5	·17	·240	1·47	3·67	·194	1·84	·222	1·58	·363
4 × 3	9½	·22	·336	2·80	7·53	1·28	3·76	·854	1·64	·677
4¾ × 1¾	6½	·18	·323	1·91	6·77	·263	2·85	·300	1·88	·371
5 × 3	11	·22	·376	3·24	13·6	1·46	5·45	·974	2·05	·672
5 × 4½	18	·29	·448	5·29	22·7	5·66	9·08	2·51	2·07	1·03
6 × 3	12	·26	·348	3·53	20·2	1·34	6·74	·892	2·40	·616
6 × 4½	20	·37	·431	5·88	34·7	5·41	11·6	2·40	2·43	·959
6 × 5	25	·41	·520	7·35	43·6	9·11	14·5	3·64	2·44	1·11
7 × 4	16	·25	·387	4·71	39·2	3·41	11·2	1·71	2·89	·851
8 × 4	18	·28	·402	5·30	55·7	3·57	13·9	1·79	3·24	·821
8 × 5	28	·35	·575	8·24	89·4	10·3	22·3	4·10	3·29	1·12
8 × 6	35	·44	·597	10·3	111	17·9	27·6	5·98	3·28	1·32
9 × 4	21	·30	·460	6·18	81·1	4·20	18·0	2·10	3·62	·824
9 × 7	58	·55	·924	17·1	230	46·3	51·1	13·2	3·67	1·65
10 × 5	30	·36	·552	8·82	146	9·78	29·1	3·91	4·06	1·05
10 × 6	42	·40	·736	12·4	212	22·9	42·3	7·64	4·14	1·36
10 × 8	70	·60	·970	20·6	345	71·6	69·0	17·9	4·09	1·87
12 × 5	32	·35	·550	9·41	220	9·74	36·7	3·90	4·84	1·02
12 × 6	44	·40	·717	12·9	315	22·3	52·6	7·42	4·94	1·31
12 × 6	54	·50	·883	15·9	376	28·3	62·6	9·43	4·86	1·33
14 × 6	46	·40	·698	13·5	441	21·6	62·9	7·20	5·71	1·26
14 × 6	57	·50	·873	16·8	553	27·9	76·2	9·31	5·64	1.29
15 × 5	42	·42	·647	12·4	428	11·9	57·1	4·78	5·89	·983
15 × 6	59	·50	·830	17·3	629	28·2	83·9	9·40	6·02	1·28
16 × 6	62	·55	·847	18·2	726	27·1	90·7	9·02	6·31	1·22
18 × 7	75	·55	·928	22·1	1150	46·6	128	13·3	7·22	1·45
20 × 7½	89	·60	1·01	26·2	1671	62·6	167	16·7	7·99	1·55
24 × 7½	100	·60	1·07	29·4	2655	66·9	221	17·8	9·50	1·51

* Published by permission of the Engineering Standards Committee. The Tables of British Standard I Beams, Channels, and Zed Bars are reprinted from Report No. 6 as issued by the Committee. Additional calculations have been inserted in the Tables of British Standards Unequal Angles, Equal Angles, and Tee Bars for thicknesses other than those calculated by the Committee, such calculations having been taken by permission from the *Pocket Companion* issued by Messrs. Dorman, Long & Co., Ltd.

TABLE 18.—PROPERTIES OF BRITISH STANDARD CHANNELS.

Size. A × B.	Standard Thicknesses.		Wt. per Foot.	Area.	Dimension P.	Moments of Inertia.		Section Moduli.		Radii of Gyration.	
	t.	T.				About XX.	About YY.	About XX.	About YY.	About XX.	About YY.
Ins.	Ins.	Ins.	Lbs.	Sq. ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
15×4	·525	·630	41·94	12·334	·935	377·0	14·55	50·27	4·748	5·53	1·09
12×4	·525	·625	36·47	10·727	1·031	218·2	13·65	36·36	4·599	4·51	1·13
12×3½	·500	·600	32·88	9·671	·867	190·7	8·922	31·79	3·389	4·44	·960
12×3½	·375	·500	26·10	7·675	·860	158·6	7·572	26·44	2·868	4·55	·993
11×3½	·475	·575	29·82	8·771	·896	148·6	8·421	27·02	3·234	4·12	·980
10×4	·475	·575	30·16	8·871	1·102	130·7	12·02	26·14	4·147	3·84	1·164
10×3½	·475	·575	28·21	8·296	·933	117·9	8·194	23·59	3·192	3·77	·99
10×3½	·375	·500	23·55	6·925	·933	102·6	7·187	20·52	2·800	3·85	1·02
9×3½	·450	·550	25·39	7·469	·971	88·07	7·660	19·57	3·029	3·43	1·01
9×3½	·375	·500	22·27	6·550	·976	79·90	6·963	17·76	2·759	3·49	1·03
9×3	·375	·437	19·37	5·696	·754	65·18	4·021	14·48	1·790	3·38	·840
8×3½	·425	·525	22·72	6·682	1·011	63·76	7·067	15·94	2·839	3·09	1·03
8×3	·375	·500	19·30	5·675	·844	53·43	4·329	13·36	2·008	3·07	·873
7×3½	·400	·500	20·23	5·950	1·061	44·55	6·498	12·73	2·664	2·74	1·04
7×3	·375	·475	17·56	5·166	·874	37·63	4·017	10·75	1·829	2·70	·882
6×3½	·375	·475	17·9	5·266	1·119	29·66	5·907	9·885	2·481	2·36	1·06
6×3	·312	·437	14·49	4·261	·938	24·01	3·503	8·003	1·699	2·37	·907

TABLE 19.—PROPERTIES OF BRITISH STANDARD Z BARS.

Size. A×B.	Standard Thicknesses.		Area.	Wt. per Foot.	Moments of Inertia.		Section Moduli.		Angle *a* in degrees.	Least radius of Gyration
	t.	T.			About XX.	About YY.	About XX.	About YY.		
Ins.	Ins.	Ins.	Sq. ins.	Lbs.	Ins.	Ins.	Ins.	Ins.		Ins.
10×3½	·475	·575	8·283	28·16	117·865	12·876	23·573	3·947	14	·839
9×3½	·450	·550	7·449	25·33	87·889	12·418	19·531	3·792	16½	·843
8×3½	·425	·525	6·670	22·68	63·729	12·024	15·932	3·657	19½	·845
7×3½	·400	·500	5·948	20·22	44·609	11·618	12·745	3·521	23	·840
6×3½	·375	·475	5·258	17·88	29·660	11·134	9·887	3·361	28½	·821
5×3	·350	·450	4·169	14·17	16·145	6·578	6·458	2·328	29½	·698

TABLE 20.—PROPERTIES OF BRITISH STANDARD UNEQUAL ANGLES.

Size and Thickness.	Area.	Wt. per Foot.	Dimensions.		Moments of Inertia.		Section Moduli.		Angle *a* in degrees.	Least Radius of Gyration.
			J.	P.	About XX.	About YY.	About XX.	About YY.		
Ins.	Sq.ins.	Lbs.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.		Ins.
7 ×3½× ½	5·0	17·00	2·50	·764	25·1	4·28	5·58	1·56	14½	·74
7 ×3½+ ⅝	6·172	20·98	2·55	·814	30·55	5·15	6·86	1·92	14½	·74
7 ×3½× ¾	7·313	24·86	2·60	·862	35·68	5·95	8·11	2·26	14	·73
6½×4½× ½	5·248	17·84	2·08	1·09	22·2	8·75	5·02	2·57	25	·97
6½×4½× ⅝	6·482	22·04	2·13	1·14	27·09	10·60	6·20	3·15	25	·96
6½×4½× ¾	7·686	26·13	2·18	1·19	31·66	12·32	7·33	3·72	25	·96
6½×3½× ⅜	3·610	12·27	2·22	·741	15·7	3·27	3·67	1·18	16½	·75
6½×3½× ½	4·750	16·15	2·28	·792	20·4	4·20	4·83	1·55	16½	·75
6½×3½× ⅝	5·860	19·92	2·33	·841	24·83	5·06	5·95	1·90	16	·74
6 ×4 × ⅜	3·610	12·27	1·91	·923	13·2	4·73	3·23	1·54	23½	·87
6 ×4 × ½	4·750	16·15	1·96	·974	17·1	6·10	4·23	2·02	23½	·86
6 ×4 × ⅝	5·860	19·92	2·02	1·02	20·8	7·36	5·23	2·47	23½	·86
6 ×3½× ⅜	3·424	11·64	2·01	·773	12·6	3·22	3·16	1·18	19	·76
6 ×3½× ½	4·502	15·31	2·06	·823	16·4	4·14	4·16	1·55	19	·75
6 ×3½× ⅝	5·549	18·87	2·11	·872	19·88	4·97	5·11	1·89	18½	·75
5½×3½× ⅜	3·236	11·00	1·80	·807	9·93	3·15	2·68	1·17	22	·76
5½×3½× ½	4·252	14·46	1·85	·857	12·80	4·05	3·51	1·53	22	·75
5½×3½× ⅝	0·236	17·80	1·90	·905	15·6	4·86	4·33	1·87	21½	·75
5½×3 × ⅜	3·050	10·37	1·90	·662	9·45	2·02	2·62	·86	17	·64
5½×3 × ½	4·003	13·61	1·95	·711	12·2	2·58	3·44	1·13	16½	·64
5½×3 × ⅝	4·925	16·74	2·00	·759	14·7	3·08	4·20	1·37	16½	·63
5 ×4 × ⅜	3·236	11·00	1·51	1·01	7·96	4·53	2·28	1·52	32	·85
5 ×4 × ½	4·252	14·46	1·56	1·06	10·3	5·82	2·99	1·98	32	·84
5 ×4 × ⅝	5·236	17·80	1·60	1·11	12·4	7·01	3·66	2·43	32	·83
5 ×3½× ⅜	3·050	10·37	1·59	·848	7·64	3·09	2·24	1·17	25½	·75
5 ×3½× ½	4·003	13·61	1·64	·897	9·86	3·96	2·93	1·52	25½	·75
5 ×3½× ⅝	4·925	16·74	1·69	·944	11·9	4·75	3·60	1·86	25	·74
5 ×3 × 5/16	2·402	8·17	1·66	·667	6·14	1·68	1·84	·72	20	·65
5 ×3 × ⅜	2·859	9·72	1·68	·693	7·24	1·97	2·18	·85	19½	·65
5 ×3 × ½	3·749	12·75	1·73	·742	9·33	2·51	2·85	1·11	19½	·64
5 ×3 × ⅝	4·609	15·67	1·78	·789	11·25	3·00	3·49	1·36	19	·64

TABLE 20.—UNEQUAL ANGLES (*Continued*).

Size and Thickness.	Area.	Wt. per Foot.	Dimensions.		Moments of Inertia.		Section Moduli.		Angle α in degrees.	Least Radius of Gyration.
			J.	P.	About XX.	About YY.	About XX.	About YY.		
Ins.	Sq. ins.	Lbs.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.		Ins.
4½ × 3½ × 5/16	2·402	8·17	1·36	·866	4·22	2·55	1·54	·97	30½	·74
4½ × 3½ × 3/8	2·859	9·72	1·39	·891	5·69	3·00	1·83	1·15	30½	·74
4½ × 3½ × ½	3·749	12·75	1·44	·940	7·31	3·84	2·39	1·5	30	·74
4½ × 3½ × 5/8	4·609	15·67	1·48	·987	8·81	4·61	2·92	1·83	30	·74
4 × 3½ × 5/16	2·246	7·64	1·16	·915	3·46	2·47	1·22	·96	37	·72
4 × 3½ × 3/8	2·671	9·08	1·19	·941	4·08	2·90	1·45	1·13	37	·72
4 × 3½ × ½	3·499	11·90	1·24	·990	5·23	3·71	1·89	1·48	37	·71
4 × 3½ × 5/8	4·295	14·61	1·28	1·04	6·28	4·44	2·31	1·80	36½	·71
4 × 3 × 5/16	2·091	7·11	1·24	·746	3·31	1·59	1·20	·71	28½	·64
4 × 3 × 3/8	2·485	8·45	1·27	·771	3·89	1·87	1·42	·84	28½	·64
4 × 3 × ½	3·251	11·05	1·31	·819	4·98	2·37	1·85	1·09	28½	·63
4 × 3 × 5/8	3·985	13·55	1·36	·865	5·96	2·83	2·26	1·33	28	·63
3½ × 3 × 5/16	1·934	6·58	1·04	·792	2·27	1·53	·92	·69	35½	·62
3½ × 3 × 3/8	2·298	7·81	1·07	·819	2·67	1·80	1·10	·83	35½	·62
3½ × 3 × ½	3·001	10·20	1·11	·867	3·40	2·28	1·42	1·07	35½	·61
3½ × 3 × 5/8	3·673	12·49	1·16	·912	4·05	2·71	1·73	1·30	35	·61
3½ × 2½ × 5/16	1·799	6·05	1·12	·627	2·15	·910	·90	·49	26½	·54
3½ × 2½ × 3/8	2·111	7·18	1·15	·652	2·52	1·06	1·07	·57	26	·53
3½ × 2½ × ½	2·752	9·36	1·20	·699	3·20	1·34	1·39	·74	26	·53
3 × 2½ × ¼	1·312	4·46	·895	·648	1·14	·716	·54	·39	34	·52
3 × 2½ × 3/8	1·921	6·53	·945	·697	1·62	1·02	·79	·57	34	·52
3 × 2½ × ½	2·499	8·50	·992	·744	2·05	1·28	1·02	·73	33½	·52
3 × 2 × ¼	1·187	4·04	·976	·482	1·06	·373	·52	·25	23½	·43
3 × 2 × 3/8	1·733	5·89	1·03	·532	1·50	·525	·76	·36	23	·42
3 × 2 × ½	2·249	7·65	1·07	·578	1·89	·656	·98	·46	22½	·42
2½ × 2 × ¼	1·063	3·61	·774	·527	·636	·359	·37	·24	32	·42
2½ × 2 × 5/16	1·309	4·45	·799	·552	·770	·433	·45	·30	31½	·42
2½ × 2 × 3/8	1·547	5·26	·823	·575	·895	·502	·53	·35	31½	·42
2 × 1½ × 3/16	·622	2·11	·627	·381	·240	·115	·17	·10	28½	·32
2 × 1½ × ¼	·814	2·77	·653	·407	·308	·146	·23	·13	28	·31
2 × 1½ × 5/16	·997	3·39	·678	·431	·369	·174	·28	·16	28	·31

TABLE 21.—BRITISH STANDARD EQUAL ANGLES.

Sizes.	Area.	Weight per Foot.	J.	I_{XX}.	Section Modulus about XX.	Least Radius of Gyration
Inches.	Sq. Ins.	Lbs.	Inches.	Inches.	Inches.	Inches.
8 × 8 × ½	7·75	26·35	2·15	47·4	8·10	1·58
8 × 8 × ⅝	9·61	32·67	2·20	58·2	10·03	1·57
8 × 8 × ¾	11·44	38·89	2·25	68·5	11·91	1·56
6 × 6 × 7/16	5·06	17·21	1·64	17·3	3·97	1·18
6 × 6 × ⅝	7·11	24·18	1·71	23·8	5·55	1·18
6 × 6 × ¾	8·44	28·70	1·76	27·8	6·56	1·17
5 × 5 × ⅝	3·61	12·27	1·37	8·51	2·24	·98
5 × 5 × ½	4·75	16·15	1·42	11·0	3·07	·98
5 × 5 × ⅝	5·86	19·92	1·47	13·4	3·80	·98
4½ × 4½ × ⅜	3·24	11·00	1·22	6·14	1·87	·88
4½ × 4 × ½ ½	4·25	14·46	1·29	7·92	2·47	·87
4½ × 4½ × ⅝	5·24	17·80	1·34	9·56	3·03	·87
4 × 4 × ⅜	2·86	9·72	1·12	4·26	1·48	·78
4 × 4 × ½	3·75	12·75	1·17	5·46	1·93	·77
4 × 4 × ⅝	4·61	15·67	1·22	6·56	2·36	·77
3½ × 3½ × 5/16	2·09	7·11	·97	2·39	·95	·68
3½ × 3½ × ⅜	2·48	8·45	1·00	2·80	1·12	·68
3½ × 3½ × ½	3·25	11·05	1·05	3·57	1·46	·68
3½ × 3¼ × ⅝	3·98	13·55	1·09	4·27	1·77	·68
3 × 3 × ¼	1·44	4·90	·827	1·21	·56	·59
3 × 3 × ⅜	2·11	7·18	·877	1·72	·81	·58
3 × 3 × ½	2·75	9·36	·924	2·19	1·05	·58
3 × 3 × ⅝	3·36	11·43	·970	2·59	1·28	·58
2½ × 2½ × ¼	1·19	4·04	·703	·677	·38	·48
2½ × 2½ × 5/16	1·46	4·98	·728	·822	·46	·48
2½ × 2½ × ⅜	1·73	5·89	·752	·962	·55	·48
2½ × 2½ × ½	2·25	7·65	·799	1·21	·71	·48
2¼ × 2¼ × 3/16	·809	2·75	·616	·378	·23	·44
2¼ × 2¼ × ¼	1·06	3·61	·643	·489	·30	·44
2¼ × 2¼ × 5/16	1·31	4·45	·668	·592	·37	·43
2¼ × 2¼ × ⅜	1·55	5·26	·692	·686	·44	·43
2 × 2 × 3/16	·715	2·43	·554	·260	·18	·39
2 × 2 × ¼	·938	3·19	·581	·336	·24	·39
2 × 2 × 5/16	1·15	3·92	·605	·401	·29	·38
2 × 2 × ⅜	1·36	4·62	·629	·467	·34	·38
1¾ × 1¾ × 3/16	·622	2·11	·495	·172	·14	·34
1¾ × 1¾ × ¼	·814	2·77	·520	·220	·18	·34
1¾ × 1¾ × 5/16	·997	3·39	·544	·264	·22	·34
1½ × 1½ × 3/16	·526	1·79	·434	·105	·10	·29
1½ × 1½ × ¼	·686	2·33	·458	·134	·13	·29
1½ × 1½ × 5/16	·839	2·85	·482	·159	·16	·29
1¼ × 1¼ × 3/16	·433	1·47	·371	·058	·07	·24
1¼ × 1¼ × ¼	·561	1·91	·396	·073	·09	·23

TABLE 22.—BRITISH STANDARD TEES.

Sizes.	Area.	Wt. per Foot.	J.	Moments of Inertia.		Section Moduli.		Radii of Gyration.	
				XX.	YY.	XX.	YY.	XX.	YY.
Ins.	Sq. ins.	Lbs.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.	Ins.
6 × 4 × ⅜	3·634	12·36	·915	4·70	6·34	1·52	2·11	1·14	1·32
6 × 4 × ½	4·771	16·22	·968	6·07	8·62	2·00	2·87	1·13	1·34
6 × 4 × ⅝	5·878	19·99	1·02	7·35	10·91	2·47	3·64	1·12	1·36
6 × 3 × ⅜	3·260	11·08	·633	2·06	6·39	·87	2·13	·795	1·40
6 × 3 × ½	4·272	14·53	·684	2·63	8·65	1·14	2·88	·785	1·42
6 × 3 × ⅝	5·256	17·87	·732	3·14	10·94	1·39	3·65	·773	1·44
5 × 4 × ⅜	3·257	11·07	·998	4·47	3·69	1·49	1·48	1·17	1·06
5 × 4 × ½	4·268	14·51	1·05	5·77	5·02	1·96	2·01	1·16	1·08
5 × 3 × ⅜	2·875	9·78	·691	1·97	3·71	·85	1·49	·828	1·14
5 × 3 × ½	3·762	12·79	·741	2·52	5·03	1·11	2·01	·818	1·16
4 × 4 × ⅜	2·872	9·77	1·11	4·19	1·90	1·45	·95	1·21	·814
4 × 4 × ½	3·758	12·78	1·16	5·40	2·59	1·90	1·29	1·20	·830
4 × 3 × ⅜	2·498	8·49	·767	1·86	1·91	·83	·96	·863	·875
4 × 3 × ½	3·262	11·08	·816	2·36	2·60	1·08	1·30	·851	·893
3½ × 3½ × ⅜	2·496	8·49	·98	2·79	1·28	1·10	·73	1·05	·717
3½ × 3½ × ½	3·259	11·08	1·04	3·54	1·75	1·44	1·00	1·04	·733
3 × 3 × ⅜	2·121	7·21	·868	1·70	·816	·80	·54	·897	·620
3 × 3 × ½	2·760	9·38	·918	2·16	1·11	1·04	·74	·886	·636
3 × 2½ × ⅜	1·929	6·56	·695	1·01	·814	·56	·45	·725	·650
3 × 2½ × ½	2·506	8·52	·742	1·28	1·28	·73	·74	·713	·665
2½ × 2½ × ¼	1·197	4·07	·697	·677	·302	·38	·24	·752	·502
2½ × 2½ × 5/16	1·474	5·01	·724	·832	·387	·46	·31	·747	·512
2½ × 2½ × ⅜	1·742	5·92	·750	·959	·473	·55	·38	·742	·521
2¼ × 2¼ × ¼	1·071	3·64	·638	·488	·224	·30	·20	·675	·457
2¼ × 2¼ × ⅜	1·554	5·28	·689	·685	·349	·44	·31	·664	·474
2 × 2 × ¼	·947	3·22	·579	·337	·157	·24	·16	·597	·407
2 × 2 × ⅜	1·367	4·64	·628	·469	·246	·34	·25	·586	·424
1½ × 2 × ¼	·820	2·79	·648	·307	·068	·23	·09	·612	·288
1½ × 2 × 5/16	1·003	3·41	·674	·369	·088	·28	·12	·607	·296
1¾ × 1¾ × ¼	·820	2·79	·519	·221	·107	·18	·12	·520	·361
1¾ × 1¾ × 5/16	·999	3·40	·544	·265	·137	·22	·16	·515	·370
1½ × 1½ × 3/16	·531	1·81	·435	·106	·048	·10	·06	·447	·301
1½ × 1½ × ¼	·692	2·35	·460	·135	·067	·13	·09	·442	·312

DYNAMICS.

The *Velocity* of a body at any point in its motion is the rate at which it moves at that point.

When the velocity is constant over a given time interval, the body is said to have *uniform* velocity during the interval.

Velocity may change in direction or in magnitude; the rate at which the velocity changes is called *acceleration.*

UNIFORM VELOCITY.

If v is the velocity in feet per second,
t is time in seconds,
and s is the space covered,
Then $s = v\ t.$

UNIFORM ANGULAR VELOCITY.

If ω is the angular velocity in radians per second,
N is the number of revolutions per minute,
t is the time in seconds,
θ is the complete angle (turned in time t) in radians,
r is the radius of any point on the rotating body,
and v the velocity in feet per second at that point;

$$\text{Then } \omega = \frac{2\pi N}{60}$$

$$\theta = \omega t$$

$$\text{Also } v = r\omega$$

ACCELERATION (UNIFORM).

Let v feet per second be the velocity at any moment;
And u feet per second be the velocity t seconds later.
If f feet per second be the acceleration,
And s the space in feet covered in time t,

$$\frac{v-u}{t} = f$$

$$s = ut + \tfrac{1}{2} ft.^2$$

If the initial velocity u is 0, that is, when the body starts from rest,

$$\text{Then } v = ft.$$

$$S = \tfrac{1}{2} ft.^2$$

$$\text{or } v^2 = 2fs.$$

In the case of a body falling freely in air, the acceleration due to gravity = 32·2 ft. per sec. per sec. (symbol g).

∴. A body falling from a height h acquires at the end of the fall a velocity v given by

$$v^2 = 2gh$$

ENERGY AND MOMENTUM.

The energy due to the movement of a body is called *kinetic energy* (K.E.).

A body of weight W moving with velocity v has a kinetic energy given by

$$\text{K.E.} = \frac{Wv^2}{2g}$$

The momentum of this body is given by $\frac{Wv}{g}$.

According to Newton's laws of motion force is proportional to rate of change of momentum; this gives rise to the following formula connecting the weight W of a body, the resultant force F acting on the body and the resulting acceleration a of the body in the direction of the force

$$F = \frac{Wa}{g}$$

The *potential* energy of a body with reference to a given datum line is given by the work which the body could do in moving down to this line; if W is the weight of the body and h its height above the datum, the potential energy is measured by $W \cdot h$.

The law of the *conservation of energy* states that energy cannot be destroyed but can be converted from one form to another without loss. In practice some of the energy in every machine or engine is lost as far as practical purposes are concerned because it is converted into a form (usually heat) which becomes dissipated.

Revolutions of Wheels or Disks per Minute or Wheels of given Diameter to Possess certain Peripheral Speeds.

Circumferential Velocity or Peripheral Speed.		Diameter of Wheel.								
Miles per hour.	Feet per sec.	2′	2′ 3″	2′ 6″	2′ 9″	3′	3′ 3″	3′ 6″	3′ 9″	4′
10	880	140	124	112	102	93	86	80	75	70
15	1,320	210	187	168	153	140	129	120	112	105
20	1,760	280	249	224	204	187	172	160	149	140
25	2,200	350	311	280	255	234	215	200	187	175
30	2,640	420	373	336	306	280	258	240	224	210
35	3,080	490	435	392	356	327	301	280	261	245
40	3,520	560	498	448	407	374	345	320	299	280
45	3,960	630	560	504	458	420	388	360	336	315
50	4,400	700	622	560	509	467	430	400	373	350
55	4,840	770	684	616	560	514	474	440	411	385
60	5,280	840	746	672	611	560	517	480	448	420
65	5,720	910	809	728	662	607	560	520	485	455
70	6,169	980	871	784	713	654	603	560	523	490

ROTATING BODIES.

If a body is rotating uniformly with an angular velocity ω radians per sec., or n revolutions per second, we have, as previously stated, the peripheral velocity v at radius $r = \omega r = 2\pi nr$.

Example. At what rate must the 36″ wheels of a motor-car rotate for a car speed of 45 miles per hour ? *Ans.* 420.

Centripetal Acceleration and Centrifugal Force. For a body to move in a circle with uniform velocity the velocity must be constantly changing in direction and the acceleration which represents the rate of change of the direction of the velocity is called *centripetal acceleration,* and acts towards the centre of the circle ; the outward force which tends to resist the acceleration towards the centre is called *centrifugal force.*

In the case of a body of weight W lbs. moving with a peripheral speed v feet per second or n revolutions per second at a radius r feet we have

$$\text{Centripetal acceleration} = \frac{v^2}{r} = 4\pi^2 n^2 r \text{ feet per second}$$

$$\text{Centrifugal force} = \frac{Wv^2}{gr} = \frac{4\pi^2 n^2 r W}{g} \text{ lbs.}$$

The table on page 192 gives the centrifugal force in lbs. per lb. of rotating body for various radii and revolutions per minute.

TABLE 23.—CENTRIFUGAL FORCE IN LBS. AT VARIOUS SPEEDS (PER 1 LB. OF ROTATING MASS).

RADIUS OF WHEEL PULLEY OR DISK IN INCHES.

Revolutions per Minute.	6	9	12	15	18	21	24	27	30	33	36
100	1·70	2·55	3·41	4·26	5·11	5·96	6·81	7·66	8·51	9·36	10·21
200	6·81	10·22	13·62	17·03	20·44	23·84	27·25	30·66	34·06	37·47	40·88
300	15·32	22·98	30·65	38·31	45·96	53·63	61·29	68·96	76·61	84·28	91·94
400	27·24	40·86	54·49	68·56	81·73	95·35	108·97	122·60	137·12	149·85	163·45
500	42·56	63·85	85·13	106·42	127·70	149·00	170·26	191·57	212·84	234·13	255·39
600	61·3	91·95	122·6	155·26	183·9	214·55	245·2	275·88	306·52	337·16	367·8
700	83·43	125·16	166·87	208·60	250·32	292·00	333·74	375·45	417·20	458·90	500·61
800	108·97	163·47	217·95	272·42	326·94	381·40	435·90	490·4	544·84	599·4	653·85
900	137·93	206·90	275·86	344·8	413·8	482·7	551·7	620·7	689·6	758·7	837·6
1,000	170·25	255·4	340·5	425·6	510·8	595·9	681·0	766·2	852·0	936·4	1021·5
1,100	206·0	309·0	412·0	514·9	618·0	721·0	824·0	927·0	1039·9	1132·2	1236·0
1,200	245·15	367·7	490·3	612·9	735·5	858·0	980·6	1103·3	1235·8	1348·5	1470·9
1,300	333·75	500·6	667·5	834·4	1001·3	1168·2	1335·0	1502·2	1768·3	1835·8	2002·5
1,400	435·9	653·8	871·8	1089·8	1307·6	1525·8	1743·6	1961·5	2179·6	2397·7	2615·2
1,500	551·65	827·4	1103·3	1379·0	1654·9	1931·0	2206·6	2482·8	2758·0	3034·2	3309·9
1,600	681·0	1021·6	1362·0	1543·6	2043·0	2383·4	2724·0	3065·0	3410·0	3745·5	4086·0
1,700	1064·0	1596·2	2128·0	2660·2	3192·0	3724·0	4256·0	4789·0	5320·4	5852·5	6384·0
1,800	1532·25	2298·8	3064·5	3831·05	4596·7	5362·8	6129·0	6897·0	7662·1	8427·8	9193·5

SECTION III

GENERAL ENGINEERING TABLES

TABLE 1.—WHITWORTH STANDARD BOLTS AND NUTS.

Diameter of bolt in inches.	Diameter at bottom of thread in inches.	Number of threads per inch.	Width across corners in inches.	Width across flats in inches.	Diameter of tapping holes in inches.	Thickness of head in inches.	Thickness of nut in inches.
$\frac{3}{16}$	·134	24	$\frac{1}{2}+\frac{1}{64}$	$\frac{7}{16}$	$\frac{1}{8}+\frac{1}{64}$	·164	$\frac{3}{16}$
$\frac{1}{4}$	·186	20	$\frac{9}{16}+\frac{3}{64}$	$\frac{1}{2}+\frac{1}{64}$	$\frac{3}{16}$	·219	$\frac{1}{4}$
$\frac{5}{16}$	·241	18	$\frac{11}{16}$	$\frac{9}{16}+\frac{3}{64}$	$\frac{1}{4}$	·273	$\frac{5}{16}$
$\frac{3}{8}$	·295	16	$\frac{13}{16}$	$\frac{11}{16}+\frac{1}{64}$	$\frac{1}{4}+\frac{3}{64}$	·328	$\frac{3}{8}$
$\frac{7}{16}$	·346	14	$\frac{15}{16}+\frac{1}{64}$	$\frac{13}{16}+\frac{1}{64}$	$\frac{5}{16}+\frac{3}{34}$	·383	$\frac{7}{16}$
$\frac{1}{2}$	·393	12	$1\frac{1}{16}$	$\frac{7}{8}+\frac{1}{32}$	$\frac{3}{8}+\frac{1}{32}$	·437	$\frac{1}{2}$
$\frac{9}{16}$	·456	12	$1\frac{1}{8}+\frac{1}{32}$	$1+\frac{1}{64}$	$\frac{7}{16}+\frac{1}{32}$	·492	$\frac{9}{16}$
$\frac{5}{8}$	·508	11	$1\frac{1}{4}+\frac{1}{64}$	$1+\frac{3}{32}$	$\frac{1}{2}+\frac{1}{64}$	·547	$\frac{5}{8}$
$\frac{11}{16}$	·571	11	$1\frac{3}{8}+\frac{1}{64}$	$1\frac{3}{16}+\frac{1}{64}$	$\frac{9}{16}+\frac{1}{64}$	·601	$\frac{11}{16}$
$\frac{3}{4}$	·622	10	$1\frac{1}{2}$	$1\frac{1}{4}+\frac{3}{64}$	$\frac{5}{8}$	·656	$\frac{3}{4}$
$\frac{13}{16}$	·684	10	$1\frac{9}{16}+\frac{1}{32}$	$1\frac{3}{8}+\frac{1}{64}$	$\frac{11}{16}$	·711	$\frac{13}{16}$
$\frac{7}{8}$	·733	9	$1\frac{11}{16}+\frac{1}{64}$	$1\frac{7}{16}+\frac{3}{64}$	$\frac{11}{16}+\frac{3}{64}$	·766	$\frac{7}{8}$
$\frac{15}{16}$	·795	9	$1\frac{13}{16}+\frac{1}{64}$	$1\frac{9}{16}+\frac{1}{64}$	$\frac{3}{4}+\frac{3}{64}$	·820	$\frac{15}{16}$
1	·840	8	$1\frac{15}{16}+\frac{1}{64}$	$1\frac{5}{8}+\frac{3}{64}$	$\frac{13}{16}+\frac{1}{32}$	·875	1
$1\frac{1}{8}$	·942	7	$2\frac{1}{8}+\frac{1}{32}$	$1\frac{13}{16}+\frac{3}{64}$	$\frac{15}{16}+\frac{1}{64}$	·984	$1\frac{1}{8}$
$1\frac{1}{4}$	1·067	7	$2\frac{5}{16}+\frac{3}{64}$	$2+\frac{3}{64}$	$1\frac{1}{16}$	1·094	$1\frac{1}{4}$
$1\frac{3}{8}$	1·161	6	$2\frac{1}{2}+\frac{3}{64}$	$2\frac{3}{16}+\frac{1}{32}$	$1\frac{1}{8}+\frac{3}{64}$	1·203	$1\frac{3}{8}$
$1\frac{1}{2}$	1·286	6	$2\frac{3}{4}+\frac{1}{32}$	$2\frac{3}{8}+\frac{1}{32}$	$1\frac{1}{4}+\frac{3}{64}$	1·312	$1\frac{1}{2}$
$1\frac{5}{8}$	1·369	5	$2\frac{15}{16}+\frac{1}{32}$	$2\frac{9}{16}+\frac{1}{64}$	$1\frac{3}{8}$	1·422	$1\frac{5}{8}$
$1\frac{3}{4}$	1·494	5	$3\frac{3}{16}$	$2\frac{3}{4}$	$1\frac{1}{2}$	1·531	$1\frac{3}{4}$
$1\frac{7}{8}$	1·590	$4\frac{1}{2}$	$3\frac{7}{16}+\frac{3}{64}$	$3\frac{1}{16}$	$1\frac{9}{16}+\frac{1}{32}$	1·641	$1\frac{7}{8}$
2	1·715	$4\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{1}{8}+\frac{1}{32}$	$1\frac{11}{16}+\frac{1}{32}$	1·750	2
$2\frac{1}{4}$	1·930	4	$4\frac{1}{16}+\frac{1}{32}$	$3\frac{1}{2}+\frac{3}{64}$	$1\frac{15}{16}$	1·969	$2\frac{1}{4}$
$2\frac{1}{2}$	2·180	4	$4\frac{7}{16}+\frac{3}{64}$	$3\frac{7}{8}+\frac{1}{16}$	$2\frac{3}{16}$	2·187	$2\frac{1}{2}$
$2\frac{3}{4}$	2·384	$3\frac{1}{2}$	$4\frac{13}{16}+\frac{1}{64}$	$4\frac{3}{16}$	$2\frac{3}{8}+\frac{1}{32}$	2·406	$2\frac{3}{4}$
3	2·634	$3\frac{1}{2}$	$5\frac{3}{16}+\frac{3}{64}$	$4\frac{1}{2}+\frac{1}{32}$	$2\frac{5}{8}+\frac{1}{64}$	2·625	3
$3\frac{1}{4}$	2·856	$3\frac{1}{4}$	$5\frac{9}{16}+\frac{1}{32}$	$4\frac{13}{16}+\frac{3}{64}$	$2\frac{13}{16}+\frac{3}{64}$	2·828	$3\frac{1}{4}$
$3\frac{1}{2}$	3·105	$3\frac{1}{4}$	$5\frac{15}{16}+\frac{3}{64}$	$5\frac{1}{8}+\frac{3}{64}$	$3\frac{1}{8}$	3·063	$3\frac{1}{2}$
$3\frac{3}{4}$	3·320	3	$6\frac{3}{8}+\frac{1}{32}$	$5\frac{1}{2}+\frac{3}{64}$	$3\frac{5}{16}+\frac{1}{64}$	3·281	$3\frac{3}{4}$
4	3·573	3	$6\frac{7}{8}$	$5\frac{15}{16}+\frac{1}{64}$	$3\frac{9}{16}+\frac{1}{64}$	3·500	4
$4\frac{1}{4}$	3·804	$2\frac{7}{8}$	$7\frac{5}{16}+\frac{3}{64}$	$6\frac{3}{8}$	$3\frac{3}{4}+\frac{3}{64}$	3·703	$4\frac{1}{4}$
$4\frac{1}{2}$	4·054	$2\frac{7}{8}$	$7\frac{7}{8}$	$6\frac{13}{16}+\frac{1}{64}$	$4+\frac{3}{64}$	3·938	$4\frac{1}{2}$
$4\frac{3}{4}$	4·284	$2\frac{3}{4}$	$8\frac{7}{16}$	$7\frac{1}{4}+\frac{3}{64}$	$4\frac{1}{4}+\frac{1}{32}$	4·156	$4\frac{3}{4}$
5	4·534	$2\frac{3}{4}$	$9+\frac{1}{64}$	$7\frac{3}{4}+\frac{3}{64}$	$4\frac{1}{2}+\frac{1}{32}$	4·375	5
$5\frac{1}{2}$	5·012	$2\frac{5}{8}$	$10\frac{3}{16}+\frac{1}{32}$	$8\frac{13}{16}+\frac{1}{32}$	$5+\frac{1}{64}$	4·813	$5\frac{1}{2}$
6	5·487	$2\frac{1}{2}$	$11\frac{1}{2}+\frac{3}{64}$	10	$5\frac{7}{16}+\frac{3}{64}$	5·250	6

TABLE 2.—UNITED STATES' STANDARD BOLTS AND NUTS.

Diameter of bolt in inches.	Diameter at bottom of thread in inches.	Number of threads per inch.	Width of flat summits and base of thread.	Diameter of bolt in inches.	Diameter at bottom of thread in ins.	Number of threads per inch.	Width of flat summits and base of thread.
$\frac{1}{4}$	·185	20	·0062	2	1·712	$4\frac{1}{2}$	·0277
$\frac{5}{16}$	·240	18	·0074	$2\frac{1}{4}$	1·962	$4\frac{1}{2}$	·0277
$\frac{3}{8}$	·294	16	·0078	$2\frac{1}{2}$	2·176	4	·0312
$\frac{7}{16}$	·344	14	·0089	$2\frac{3}{4}$	2·426	4	·0312
$\frac{1}{2}$	·400	13	·0096	3	2·629	$3\frac{1}{2}$	·0357
$\frac{9}{16}$	·454	12	·0104	$3\frac{1}{4}$	2·879	$3\frac{1}{2}$	·0357
$\frac{5}{8}$	·507	11	·0113	$3\frac{1}{2}$	3·100	$3\frac{1}{4}$	·0384
$\frac{3}{4}$	·620	10	·0125	$3\frac{3}{4}$	3·317	3	·0413
$\frac{7}{8}$	·731	9	·0138	4	3·567	3	·0413
1	·837	8	·0156	$4\frac{1}{4}$	3·798	$2\frac{7}{8}$	·0435
$1\frac{1}{8}$	·940	7	·0178	$4\frac{1}{2}$	4·028	$2\frac{3}{4}$	·0454
$1\frac{1}{4}$	1·065	7	·0178	$4\frac{3}{4}$	4·256	$2\frac{5}{8}$	·0476
$1\frac{3}{8}$	1·160	6	·0208	5	4·480	$2\frac{1}{2}$	·0500
$1\frac{1}{2}$	1·284	6	·0208	$5\frac{1}{4}$	4·730	$2\frac{1}{2}$	·0500
$1\frac{5}{8}$	1·389	$5\frac{1}{2}$	·0227	$5\frac{1}{2}$	4·953	$2\frac{3}{8}$	·0526
$1\frac{3}{4}$	1·491	5	·0250	$5\frac{3}{4}$	5·203	$2\frac{3}{8}$	·0526
$1\frac{7}{8}$	1·616	5	·0250	6	5·423	$2\frac{1}{4}$	·0555

Thickness of head = diameter of bolt − $\frac{1}{16}$th.
Breadth of nut across flats = $1\frac{1}{2}$ diameter + 0·1.

TABLE 3.—FRENCH STANDARD BOLTS AND NUTS.

Diameter of screw.		Diameter of bottom of thread.		Pitch of thread.	Number of threads per inch.	Width across flats.		Width across corners.		Thickness of head.	
Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.		Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.
5	·196	3·30	·13	1·4	18·14	13·97	·55	16·0	·63	5	·196
7·5	·295	5·58	·22	1·6	15·87	17·27	·68	19·81	·78	7	·275
10	·393	7·87	·31	1·8	14·11	22·35	·88	25·90	1·02	9	·355
12·5	·492	9·90	·39	2	12·70	26·42	1·04	30·48	1·20	11	·433
15	·589	12·19	·48	2·2	11·55	30·48	1·20	35·05	1·38	13	·51
17·5	·688	14·73	·58	2·4	10·58	34·54	1·36	39·87	1·57	15	·59
20	·787	16·76	·66	2·6	9·77	38·60	1·52	44·45	1·75	17	·67
22·5	·885	19·20	·76	2·8	9·07	42·67	1·68	49·27	1·94	19	·75
25	·984	21·34	·84	3	8·46	46·74	1·84	53·84	2·12	21	·83
30	1·178	25·90	1·02	3·4	7·47	54·86	2·16	63·50	2·50	24	·94
35	1·376	30·48	1·20	3·8	6·68	62·99	2·48	72·64	2·86	27	1·06
40	1·574	35·56	1·40	4·2	6·05	71·12	2·80	82·04	3·23	31	1·22
45	1·770	39·62	1·56	4·6	5·52	79·25	3·12	91·44	3·60	35	1·38
50	1·968	44·19	1·74	5	5·08	87·37	3·44	100·80	3·97	38	1·50
55	2·162	48·76	1·92	5·4	4·70	95·50	3·76	110·2	4·34	41	1·65
60	2·356	52·83	2·08	5·8	4·38	103·63	4·08	119·8	4·72	44	1·73
65	2·554	57·42	2·26	6·2	4·09	111·76	4·40	129·0	5·08	48	1·89
70	2·752	61·97	2·44	6·6	3·85	118·88	4·72	138·4	5·45	52	2·05
75	2·950	66·04	2·60	7·0	3·63	128·02	5·04	147·8	5·82	56	2·20
80	3·149	70·61	2·78	7·4	3·43	136·14	5·36	157·5	6·20	60	2·36

TABLE 4.—WHITWORTH'S GAS THREADS.

Bore of pipe in inches.	Exterior diameter of pipe in inches.	Exterior diameter of pipe in decimals.	Number of threads per inch.	Diameter at bottom of thread in decimals.
$\frac{1}{8}$	$\frac{3}{8}$	·3825	28	·3367
$\frac{1}{4}$	$\frac{1}{2} + \frac{1}{64}$	·5180	19	·4506
$\frac{3}{8}$	$\frac{5}{8} + \frac{1}{32}$	·6563	19	·5889
$\frac{1}{2}$	$\frac{13}{16} + \frac{1}{64}$	·8200	14	·7342
$\frac{5}{8}$	$\frac{7}{8} + \frac{1}{64}$	·9022	14	·8107
$\frac{3}{4}$	$1\frac{1}{32} + \frac{1}{64}$	1·041	14	·9495
$\frac{7}{8}$	$1\frac{3}{16}$	1·189	14	1·0975
1	$1\frac{5}{16}$	1·309	11	1·1925
$1\frac{1}{8}$	$1\frac{1}{2}$	1·492	,,	1·3755
$1\frac{1}{4}$	$1\frac{5}{8} + \frac{1}{64}$	1·650	,,	1·5335
$1\frac{3}{8}$	$1\frac{3}{4}$	1·745	,,	1·6285
$1\frac{1}{2}$	$1\frac{7}{8}$	1·883	,,	1·7660
$1\frac{5}{8}$	2	2·021	,,	1·9045
$1\frac{3}{4}$	$2\frac{3}{64}$	2·047	,,	1·9305
$1\frac{7}{8}$	$2\frac{1}{4}$	2·245	,,	2·1285
2	$2\frac{5}{16} + \frac{1}{32}$	2·347	,,	2·2305
$2\frac{1}{8}$	$2\frac{7}{16} + \frac{1}{32}$	2·460	,,	2·3435
$2\frac{1}{4}$	$2\frac{9}{16} + \frac{1}{32}$	2·588	,,	2·4710
$2\frac{3}{8}$	$2\frac{3}{4} + \frac{3}{64}$	2·790	,,	2·6735
$2\frac{1}{2}$	3	3·001	,,	2·8848
$2\frac{5}{8}$	$3\frac{1}{8}$	3·120	,,	3·0035
$2\frac{3}{4}$	$3\frac{1}{4}$	3·247	,,	3·1305
$2\frac{7}{8}$	$3\frac{5}{16} + \frac{3}{64}$	3·360	,,	3·2435
3	$3\frac{7}{16} + \frac{3}{64}$	3·485	,,	3·3685
$3\frac{1}{4}$	$3\frac{11}{16} + \frac{1}{32}$	3·699	,,	3·5820
$3\frac{1}{2}$	$3\frac{7}{8} + \frac{1}{32}$	3·912	,,	3·7955
$3\frac{3}{8}$	$4\frac{1}{8}$	4·126	,,	4·0090
4	$4\frac{5}{16} + \frac{1}{32}$	4·339	,,	4·2225

TABLE 5.—BRITISH ASSOCIATION (B.A.) SCREW THREADS.

NUMBER.	THREADS PER INCH.	ABSOLUTE DIMENSIONS IN MILLIMETRES.		EQUIVALENT DIMENSIONS THOUSANDTHS OF AN INCH (*Mils*.).	
		Diameter.	Pitch.	Diameter.	Pitch.
25	353	0·25	0·07	10	2·3
24	317	0·29	0·08	11	3·1
23	285	0·33	0·09	13	3·5
22	259	0·37	0·10	15	3·9
21	231	0·42	0·11	17	4·3
20	212	0·48	0·12	19	4·7
19	181	0·54	0·14	21	5·5
18	169	0·62	0·15	24	5·9
17	149	0·70	0·17	28	6·7
16	134	0·79	0·19	31	7·5
15	121	0·90	0·21	35	8·3
14	110	1·00	0·23	39	9·1
13	101	1·20	0·25	47	9·8
12	90·7	1·30	0·28	51	11·0
11	81·9	1·50	0·31	59	12·2
10	72·6	1·70	0·35	67	12·8
9	65·1	1·90	0·39	75	15·4
8	59·1	2·20	0·43	87	16·9
7	52·9	2·50	0·48	98	18·9
6	47·9	2·80	0·53	110	20·9
5	43·0	3·20	0·59	126	23·2
4	38·5	3·60	0·66	142	26·0
3	34·8	4·10	0·73	161	28·7
2	31·4	4·70	0·81	185	31·9
1	28·2	5·30	0·90	200	35·4
0	25·4	6·00	1·00	256	39·4

Note. Angle of thread = 47½°.
Rounded off top and bottom $\frac{1}{5}$.

TABLE 6.—STRENGTH OF B.A. AND B.S.F. BOLTS.

Diameter (ins.).		Cross section area (ins.)^2.		*MI* for bending (on full diameter), ins.^{4}	*Z* for bending (on full diameter), ins.^{3}	Allowable B.M. (lbs. ins.) for stress of 80,000 lbs./ins.^{2}	Allowable shear load on stress of 40,000 lbs./ins.^{2}	Allowable tensile load on stress of 80,000 lbs./ins.^{2}
Full.	Core.	Full diameter.	Core diameter B.S.F. thread.					
4BA ·142 ins.	·11	·0158	·0096	$·1985 \times 10^{-4}$	$·2800 \times 10^{-3}$	22	630	780
·2BA ·185 ins.	·147	·0269	·0169	$·5770 \times 10^{-4}$	$·6240 \times 10^{-3}$	51	1080	1350
$\frac{1}{4}$,,	·1988	·0491	·0316	$·1917 \times 10^{-3}$	$·1534 \times 10^{-2}$	123	1960	2530
$\frac{5}{16}$,,	·2543	·0767	·0508	$·4675 \times 10^{-3}$	$·299 \times 10^{-2}$	240	3070	4070
$\frac{3}{8}$,,	·3110	·1105	·0760	$·9707 \times 10^{-3}$	$·5177 \times 10^{-2}$	414	4430	6080
$\frac{7}{16}$,,	·3664	·1503	·1054	$·1800 \times 10^{-2}$	$·8220 \times 10^{-2}$	658	6000	8450
$\frac{1}{2}$,,	·4200	·1954	·1385	$·3068 \times 10^{-2}$	$·1227 \times 10^{-1}$	982	7850	11100
$\frac{9}{16}$,,	·4825	·2485	·1828	$·4920 \times 10^{-2}$	$·1750 \times 10^{-1}$	1400	9950	14600
$\frac{5}{8}$,,	·5335	·3068	·2235	$·7490 \times 10^{-2}$	$·2397 \times 10^{-1}$	1920	12250	17900
$\frac{11}{16}$,,	·5960	·3712	·2790	$·1094 \times 10^{-1}$	$·3185 \times 10^{-1}$	2550	14850	22300
$\frac{3}{4}$,,	·6433	·4418	·3250	$·1553 \times 10^{-1}$	$·4142 \times 10^{-1}$	3320	17700	26000
$\frac{13}{16}$,,	·7058	·5185	·3913	$·2140 \times 10^{-1}$	$·5270 \times 10^{-1}$	4210	20700	31300
$\frac{7}{8}$,,	·7586	·6013	·4520	$·2877 \times 10^{-1}$	$·6577 \times 10^{-1}$	5260	24000	36200
$\frac{15}{16}$,,	—	·6903	·5295	$·3800 \times 10^{-1}$	$·8110 \times 10^{-1}$	6490	27600	42300
1 ,,	·8719	·7854	·5971	$·4909 \times 10^{-1}$	$·9817 \times 10^{-1}$	7860	31400	47800
$1\frac{1}{8}$,,	·9827	·9940	·7585	$·7863 \times 10^{-1}$	·1398	11180	39600	60600
$1\frac{1}{4}$,,	1·1077	1·2272	·9637	·1198	·1917	15350	49000	77100
$1\frac{3}{8}$,,	1·2149	1·4849	1·1593	·1755	·2552	20400	59400	92600
$1\frac{7}{16}$,,	—	1·6230	—	·2100	·2920	23350	—	—
$1\frac{1}{2}$,,	1·399	1·7671	1·4100	·2485	·3313	26500	70600	113000
$1\frac{9}{16}$,,	—	1·9175	—	·2940	·3770	30100	—	—
$1\frac{5}{8}$,,	1·4649	2·0739	1·6854	·3431	·4213	33700	82800	135000
$1\frac{11}{16}$,,	—	2·2365	—	·3980	·4710	37700	—	—
$1\frac{3}{4}$,,	1·567	2·4053	1·9285	·4604	·5261	42100	96200	154000
$\frac{13}{16}$,,	—	2·5802	—	·5310	·5860	46900	—	—
$1\frac{7}{8}$,,	—	2·7612	—	·6067	·6471	51750	—	—
$1\frac{15}{16}$,,	—	2·9483	—	·6900	·7120	56900	—	—
2 ,,	1·817	3·1416	2·593	·7854	·7854	62800	125700	207000

TABLE 7.—STRENGTH OF WHITWORTH BOLTS.

Diameter of bolt in inches.	Area at bottom of thread in inches	Safe load on steel in lbs.	Safe load on wrought-iron in lbs.	Diameter of bolt in inches.	Area at bottom of thread in inches.	Safe load on steel in lbs.	Safe load on wrought-iron in lbs.
$\frac{3}{16}$	·014	94·	62·	$1\frac{5}{8}$	1·472	9891	6594
$\frac{1}{4}$	·027	181	120	$1\frac{3}{4}$	1·753	11780	7853
$\frac{5}{16}$	·046	272	206	$1\frac{7}{8}$	1·986	13345	8897
$\frac{3}{8}$	·068	456	304	2	2·311	15529	10353
$\frac{7}{16}$	·094	631	421	$2\frac{1}{4}$	2·926	19662	13108
$\frac{1}{2}$	·121	813	542	$2\frac{1}{2}$	3·733	25085	16723
$\frac{9}{16}$	·155	1041	694	$2\frac{3}{4}$	4·464	29998	19998
$\frac{5}{8}$	·204	1370	913	3	5·450	36624	24416
$\frac{11}{16}$	·256	1720	1146	$3\frac{1}{4}$	6·402	43021	28680
$\frac{3}{4}$	·304	2042	1361	$3\frac{1}{2}$	7·563	50823	33882
$\frac{13}{16}$	·367	2466	1644	$3\frac{3}{4}$	8·673	58282	38855
$\frac{7}{8}$	·422	2835	1890	4	10·027	67381	44920
$\frac{15}{16}$	·496	3313	2222	$4\frac{1}{4}$	11·365	76362	50915
1	·554	3722	2481	$4\frac{1}{2}$	12·908	86741	57827
$1\frac{1}{8}$	·697	4683	3122	$4\frac{3}{4}$	14·404	96714	64529
$1\frac{1}{4}$	·894	5907	4005	5	16·146	108501	72334
$1\frac{3}{8}$	1·058	7109	4739	$5\frac{1}{2}$	19·720	132518	88245
$1\frac{1}{2}$	1·299	8729	5819	6	23·640	158860	105907

TABLE 8.—IMPERIAL STANDARD WIRE GAUGE

Descriptive number.	Equivalent in parts of an inch.	Metric equivalent millimetres.	Sectional area of wire.	
			Square inches.	Square millimetres.
7/0	·500	12·700	·1963	126·67
6/0	·464	11·785	·1691	109·09
5/0	·432	10·973	·1466	94·56
4/0	·400	10·160	·1257	81·07
3/0	·372	9·449	·1087	70·12
2/0	·348	8·839	·0951	61·36
0	·324	8·229	·0824	53·19
1	·300	7·620	·0707	45·60
2	·276	7·010	·0598	38·58
3	·252	6·401	·0499	32·18
4	·232	5·893	·0423	27·27
5	·212	5·385	·0353	22·77
6	·192	4·877	·0289	18·68
7	·176	4·470	·0243	15·70
8	·160	4·064	·0201	12·97
9	·144	3·658	·0163	10·51
10	·128	3·251	·0129	8·30
11	·116	2·946	·0106	6·82
12	·104	2·642	·00849	5·48
13	·092	2·337	·00665	4·29
14	·080	2·032	·00503	3·24
15	·072	1·829	·00407	2·63
16	·064	1·626	·00322	2·07
17	·056	1·422	·00246	1·59
18	·048	1·219	·00181	1·17
19	·040	1·016	·00126	·811
20	·036	·914	·00102	·657
21	·032	·813	·00804	·519
22	·028	·711	·000616	·397
23	·024	·610	·000452	·292
24	·022	·559	·000380	·245
25	·020	·508	·000314	·203
26	·018	·457	·000254	·164
27	·0164	·4166	·000211	·136
28	·0148	·3759	·000173	·111
29	·0136	·3454	·000145	·0937
30	·0124	·3150	·000121	·0779
31	·0116	·2946	·000106	·0682
32	·0108	·2743	·0000916	·0591
33	·0100	·2540	·0000785	·0507

TABLE 8 (*contd.*).—IMPERIAL STANDARD WIRE GAUGE.

Descriptive number.	Equivalent in parts of an inch.	Metric equivalent millimètres.	Sectional area of wire.	
			Square inches.	Square millimetres.
34	·0092	·2337	·0000665	·0429
35	·0984	·2134	·0000554	·0357
36	·0076	·1930	·0000454	·0293
37	·0068	·1727	·0000363	·0234
38	·0069	·1524	·0000283	·0182
39	·0052	·1321	·0000212	·0137
40	·0048	·1219	·0000181	·0187
41	·0044	·1118	·0000152	·00982
42	·0040	·1016	·0000126	·00811
43	·0036	·0914	·0000102	·00656
44	·0032	·0813	·00000804	·00519
45	·0028	·0711	·00000616	·00397
46	·0024	·0610	·00000452	·00292
47	·0020	·0508	·00000314	·00203
48	·0016	·0406	·00000201	·00129
49	·0012	·0305	·00000113	·00073
50	·0010	·0254	·000000785	·00051

TABLE 9.—BIRMINGHAM WIRE GAUGE.

Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.
4/0	·454	7	·180	17	·058	27	·016
3/0	·425	8	·165	18	·049	28	·014
2/0	·380	9	·148	19	·042	29	·013
0	·340	10	·134	20	·035	30	·012
1	·300	11	·120	21	·032	31	·010
2	·284	12	·109	22	·028	32	·009
3	·259	13	·095	23	·025	33	·008
4	·238	14	·083	24	·022	34	·007
5	·220	15	·072	25	·020	35	·005
6	·203	16	·065	26	·018	36	·004

TABLE 10.—WHITWORTH WIRE GAUGE.

Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.
1	·001	17	·017	55	·055	200	·200
2	·002	18	·018	60	·060	220	·220
3	·003	19	·019	65	·065	240	·240
4	·004	20	·020	70	·070	260	·260
5	·005	22	·022	75	·075	280	·280
6	·006	24	·024	80	·080	300	·300
7	·007	26	·026	85	·085	325	·325
8	·008	28	·028	90	·090	350	·350
9	·009	30	·030	95	·095	375	·375
10	·010	32	·032	100	·100	400	·400
11	·011	34	·034	110	·110	425	·425
12	·012	36	·036	120	·120	450	·450
13	·013	38	·038	135	·135	475	·475
14	·014	40	·040	150	·150	500	·500
15	·015	45	·045	165	·165		
16	·016	50	·050	180	·180		

TABLE 11.—AMERICAN STANDARD WIRE GAUGE.

(Brown and Sharpe's).

Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.
4/0	·4600	8	·1285	19	·0359	30	·01003
3/0	·4096	9	·1144	20	·0320	31	·00893
2/0	·3648	10	·1019	21	·0285	32	·00795
0	·3249	11	·0907	22	·0253	33	·00708
1	·2893	12	·0808	23	·0226	34	·00603
2	·2576	13	·0720	24	·0201	35	·00561
3	·2294	14	·0641	25	·0179	36	·00500
4	·2043	15	·0571	26	·0159	37	·00445
5	·1819	16	·0508	27	·0142	38	·00397
6	·1620	17	·0453	28	·0126	39	·00353
7	·1443	18	·0403	29	·0113	40	·00314

TABLE 12.—HOLTZAPFFEL'S LANCASHIRE GAUGE FOR ROUND STEEL WIRE AND PINION WIRE.

Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.
80	·013	51	·064	22	·152	G	·261
79	·014	50	·067	21	·157	H	·266
78	·015	49	·070	20	·160	I	·272
77	·016	48	·073	19	·164	Kj	·277
76	·018	47	·076	18	·167	K	·281
75	·019	46	·078	17	·169	L	·290
74	·022	45	·080	16	·174	M	·295
73	·023	44	·084	15	·175	N	·302
72	·024	43	·086	14	·177	O	·316
71	·026	42	·091	13	·180	P	·323
70	·027	41	·095	12	·185	Q	·332
69	·029	40	·096	11	·189	R	·339
68	·030	39	·098	10	·190	S	·348
67	·031	38	·100	9	·191	T	·358
66	·032	37	·102	8	·192	U	·368
65	·033	36	·105	7	·195	V	·377
64	·034	35	·107	6	·198	W	·386
63	·035	34	·109	5	·201	X	·397
62	·036	33	·111	4	·204	Y	·404
61	·038	32	·115	3	·209	Z	·413
60	·039	31	·118	2	·219	A1	·420
59	·040	30	·125	1	·227	B1	·431
58	·041	29	·134	A	·234	C1	·443
57	·042	28	·138	B	·238	D1	·452
56	·044	27	·141	C	·242	E1	·462
55	·050	26	·143	D	·246	F1	·475
54	·055	25	·146	E	·250	G1	·484
53	·058	24	·148	F	·257	H1	·494
52	·060	23	·150				

TABLE 13.—WARRINGTON WIRE GAUGE (Rylands Bros.).

Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.
7/0	·5	4	·229	13	·090
6/0	·46875	5	·209	14	·079
5/0	·4375	6	·191	15	·069
4/9	·40625	7	·174	16	·0625
3/0	·375	8	·159	17	·053
2/0	·34375	9	·146	18	·047
0	·326	10	·135	19	·041
1	·300	10½	·125	20	·036
2	·274	11	·117	21	·0315
3	·250	12	·100	22	·028

TABLE 14.—BIRMINGHAM SHEET-IRON GAUGE.

Descriptive number.	Size in inches.	Descriptive number.	Size in inches.	Descriptive number.	Size in inches.
1	·3125	12	·1125	23	·028125
2	·28125	13	·10	24	·025
3	·25	14	·0875	25	·02344
4	·234375	15	·075	26	·021875
5	·21875	16	·0625	27	·020312
6	·203125	17	·05625	28	·01875
7	·1875	18	·05	29	·01719
8	·171875	19	·04375	30	·015625
9	·15625	20	·0375	31	·01406
10	·140625	21	·034375	32	·0125
11	·125	22	·03125		

TABLE 15.—WEIGHTS OF STEEL BARS IN LENGTHS OF 10 FEET.

Size.	Square.	Round.	Octagon.	Hexagon.
Inches	Lbs	Lbs	Lbs	Lbs.
1/4	2·13	1·67	1·8	1·9
5/16	3·32	2·61	2·8	2·9
3/8	4·78	3·76	4·1	4·2
7/16	6·51	5·11	5·5	5·7
1/2	8·49	6·68	7·2	7·5
9/16	10·76	8·45	9·1	9·4
5/8	13·28	10·43	11·2	11·7
11/16	16·07	12·62	13·6	14·1
3/4	19·12	15·02	16·2	16·8
13/16	22·45	17·63	19·0	19·7
7/8	26·03	20·44	22·0	22·9
15/16	29·88	23·47	25·3	26·2
1	34·00	26·70	28·7	29·9
1 1/16	38·50	30·30	32·5	33·7
1 1/8	43·03	33·80	36·4	37·8
1 3/8	48·10	37·90	40·6	42·1
1 1/4	53·12	41·72	45·0	46·6
1 5/16	58·80	46·30	49·6	51·4
1 3/8	64·28	50·49	54·5	56·5
1 7/16	70·50	55·50	59·5	61·7
1 1/2	77·50	60·08	64·8	67·2
1 9/16	83·30	65·60	70·3	72·9
1 5/8	89·78	70·51	76·1	78·9
1 11/16	97·20	76·50	82·0	85·0
1 3/4	104·12	81·78	88·2	91·4
1 13/16	112·10	88·30	94·6	98·1
1 7/8	119·53	93·88	101·2	105·0
1 15/16	128·10	100·90	108·1	112·1
2	136·00	106·81	115·2	119·5
2 1/8	154·10	121·40	130·0	134·9
2 1/4	172·80	136·10	145·8	151·2
2 3/8	192·50	151·60	162·4	168·5
2 1/2	213·30	168·00	180·0	186·6
2 5/8	235·20	185·22	198·4	205·8
2 3/4	258·10	203·30	217·8	225·9
2 7/8	282·10	222·20	238·0	246·9
3	307·20	241·90	259·2	268·8

TABLE 16.—WEIGHT OF FLAT STEEL BARS PER LINEAL FOOT.

Breadth of bar in inches.	Thickness of bars in inches.														
	1/8	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	11/16	3/4	13/16	7/8	15/16	1
	Weight of bars in pounds.														
1/2	·213	·320	·425	·532	·638	·744	·851	·957	1·06	1·17	1·28	1·38	1·49	1·59	1·70
5/8	·266	·399	·532	·665	·797	·930	1·06	1·20	1·33	1·46	1·59	1·73	1·86	1·99	2·13
3/4	·319	·478	·638	·797	·957	1·12	1·28	1·43	1·59	1·75	1·91	2·07	2·23	2.39	2·55
7/8	·372	·558	·744	·930	1·11	1·30	1·49	1·67	1·86	2·05	2·23	2·42	2·60	2·79	2·98
1	·425	·638	·851	1·06	1·28	1·49	1·70	1·91	2·13	2·34	2·55	2·76	2·98	3·19	3·40
1 1/8	·478	·717	·960	1·20	1·43	1·67	1·91	2·15	2·39	2·63	2·87	3·11	3·35	3·59	3·83
1 1/4	·532	·797	1·06	1·33	1·59	1·86	2·13	2·39	2·66	2·92	3·19	3·45	3·72	3·99	4·25
1 3/8	·584	·877	1·17	1·46	1·75	2·05	2·34	2·63	2·92	3·21	3·51	3·70	4·09	4·39	4·68
1 1/2	·638	·957	1·28	1·59	1·91	2·23	2·55	2·87	3·19	3·51	3·83	4·15	4·47	4·78	5·10
1 5/8	·691	1·04	1·38	1·73	2·07	2·42	2·76	3·11	3·45	3·80	4·14	4·49	4·83	5·18	5·53
1 3/4	·744	1·11	1·49	1·86	2·23	2·60	2·98	3·35	3·72	4·09	4·46	4·84	5·21	5·58	5·96
1 7/8	·797	1·19	1·59	1·99	2·39	2·79	3·19	3·59	3·99	4·39	4·78	5·18	5·58	5·98	6·38
2	·851	1·28	1·70	2·13	2·55	2·98	3·40	3·83	4·25	4·68	5·10	5·53	5·96	6·38	6·81
2 1/8	·904	1·36	1·81	2·26	2·71	3·16	3·61	4·07	4·52	4·97	5·42	5·87	6·32	6·78	7·23
2 1/4	·958	1·43	1·91	2·39	2·87	3·35	3·83	4·31	4·78	5·26	5·74	6·22	6·79	7·18	7·66
2 3/8	1·01	1·51	2·02	2·52	3·03	3·53	4·04	4·54	5·05	5·56	6·06	6·57	7·07	7·58	8·08
2 1/2	1·06	1·59	2·13	2·66	3·19	3·72	4·25	4·78	5·31	5·86	6·38	6·92	7·44	7·98	8·51
2 5/8	1·11	1·67	2·23	2·79	3·35	3·91	4·46	5·02	5·58	6·14	6·69	7·26	7·81	8·37	8·93
2 3/4	1·16	1·75	2·34	2·92	3·51	4·09	4·68	5·26	5·84	6·43	7·01	7·60	8·18	8·77	9·36

2⅞	1·22	1·83	2·44	3·06	3·67	4·28	4·89	5·50	6·11	6·72	7·33	7·95	8·55	9·17	9·78
3	1·28	1·91	2·55	3·19	3·83	4·46	5·10	5·74	6·38	7·02	7·65	8·25	8·93	9·57	10·21
3¼	1·38	2·07	2·76	3·45	4·14	4·83	5·53	6·22	6·91	7·60	8·29	8·98	9·67	10·37	11·06
3½	1·49	2·23	2·98	3·72	4·46	5·21	5·95	6·70	7·44	8·19	8·93	9·68	10·41	11·16	11·91
3¾	1·59	2·39	3·19	3·98	4·78	5·58	6·38	7·18	7·97	8·78	9·56	10·37	11·16	11·96	12·76
4	1·70	2·55	3·40	4·25	5·10	5·95	6·80	7·66	8·50	9·36	10·20	11·06	11·90	12·76	13·68
4¼			3·61	4·51	5·44	6·32	7·23	8·13	9·04	9·94	10·85	11·75	12·65	13·56	14·46
4½			3·83	4·78	5·74	6·70	7·66	8·61	9·57	10·52	11·48	12·44	13·40	14·35	15·31
4¾			4·04	5·05	6·06	7·07	8·08	9·09	10·10	11·11	12·12	13·13	14·14	15·15	16·16
5			4·25	5·32	6·38	7·44	8·50	9·57	10·63	11·70	12·76	13·82	14·89	15·95	17·01
5½			4·68	5·84	7·02	8·19	9·36	10·53	11·70	12·87	14·04	15·21	16·37	17·54	18·71
6			5·10	6·38	7·66	8·93	10·21	11·48	12·76	14·04	15·31	16·59	17·86	19·14	20·41
6½			5·53	6·91	8·29	9·68	11·06	12·44	13·82	15·20	16·59	17·97	19·35	20·73	22·12
7			5·95	7·44	8·93	10·42	11·91	13·40	14·89	16·37	17·86	19·35	20·84	21·33	23·82
8			6·80	8·50	10·20	11·90	13·60	15·30	17·00	18·70	20·40	22·10	23·80	25·50	27·20
9			7·66	9·57	11·48	13·39	15·31	17·23	19·14	21·05	22·97	24·88	26·80	28·71	30·60
10			8·51	10·63	12·76	14·89	17·01	19·14	21·27	23·39	25·32	27·65	29·77	31·90	34·03
11			9·36	11·70	14·04	16·37	18·71	21·05	23·39	25·73	28·07	30·41	32·75	35·09	37·43
12			10·21	12·76	15·31	17·86	20·42	22·97	25·52	28·07	30·62	33·18	35·73	38·28	40·80

TABLE 17.—WEIGHT OF ROUND WROUGHT-IRON PER LINEAL FOOT.

Diameter in inches.	Weight in lbs.	Diameter in inches.	Weight in lbs.	Diameter in inches.	Weight in lbs.
$\frac{1}{8}$	0·041	$2\frac{3}{4}$	19·8	8	169·8
$\frac{3}{16}$	0·092	$2\frac{7}{8}$	21·6	$8\frac{1}{4}$	180·9
$\frac{1}{4}$	0·164			$8\frac{1}{2}$	191·8
$\frac{5}{16}$	0·256	3	23·8	$8\frac{3}{4}$	203·2
$\frac{3}{8}$	0·368	$3\frac{1}{8}$	25·9		
$\frac{7}{16}$	0·501	$3\frac{1}{4}$	28·0	9	215·0
$\frac{1}{2}$	0·654	$3\frac{3}{8}$	30·2	$9\frac{1}{4}$	227·1
		$3\frac{1}{2}$	32·5	$9\frac{1}{2}$	239·6
$\frac{9}{16}$	0·828	$3\frac{5}{8}$	34·8	$9\frac{3}{4}$	252·3
$\frac{5}{8}$	1·02	$3\frac{3}{4}$	37·3		
$\frac{11}{16}$	1·24	$3\frac{7}{8}$	39·8	10	266·3
$\frac{3}{4}$	1·47			$10\frac{1}{4}$	278·9
$\frac{13}{16}$	1·73	4	42·4	$10\frac{1}{2}$	292·7
$\frac{7}{8}$	2·00	$4\frac{1}{8}$	45·2	$10\frac{3}{4}$	306·8
$\frac{15}{16}$	2·30	$4\frac{1}{4}$	47·9		
		$4\frac{3}{8}$	50·8	11	321·2
1	2·62	$4\frac{1}{2}$	53·7	$11\frac{1}{4}$	336·0
$1\frac{1}{16}$	2·96	$4\frac{5}{8}$	56·8	$11\frac{1}{2}$	351·1
$1\frac{1}{8}$	3·31	$4\frac{3}{4}$	59·9	$11\frac{3}{4}$	366·5
$1\frac{3}{16}$	3·69	$4\frac{7}{8}$	63·1		
$1\frac{1}{4}$	4·09			12	382·2
$1\frac{5}{16}$	4·51	5	66·7	$12\frac{1}{2}$	409·0
$1\frac{3}{8}$	4·95	$5\frac{1}{8}$	69·7		
$1\frac{7}{16}$	5·40	$5\frac{1}{4}$	73·1	13	442·5
$1\frac{1}{2}$	5·89	$5\frac{3}{8}$	76·7	$13\frac{1}{2}$	477·0
		$5\frac{1}{2}$	80·3		
$1\frac{9}{16}$	6·39	$5\frac{5}{8}$	84·0	14	513·0
$1\frac{5}{8}$	6·91	$5\frac{3}{4}$	87·7	$14\frac{1}{2}$	550·0
$1\frac{11}{16}$	7·46	$5\frac{7}{8}$	91·6		
$1\frac{3}{4}$	8·03			15	589·0
$1\frac{13}{16}$	8·60	6	95·5	$15\frac{1}{2}$	629·0
$1\frac{7}{8}$	9·20	$6\frac{1}{4}$	103·7		
$1\frac{15}{16}$	9·83	$6\frac{1}{2}$	112·1	16	670·0
		$6\frac{3}{4}$	120·9	$16\frac{1}{2}$	713·0
2	10·5				
$2\frac{1}{8}$	11·8	7	130·0	17	756·0
$2\frac{1}{4}$	13·3	$7\frac{1}{4}$	139·5	$17\frac{1}{2}$	802·0
$2\frac{3}{8}$	14·8	$7\frac{1}{2}$	149·3		
$2\frac{1}{2}$	16·4	$7\frac{3}{4}$	159·4	18	848·0
$2\frac{5}{8}$	18·0			$18\frac{1}{2}$	903·0

TABLE 18.—WEIGHT OF SQUARE WROUGHT-IRON PER LINEAL FOOT.

Side of square in inches.	Weight in lbs.	Side of square in inches.	Weight in lbs.	Side of square in inches.	Weight in lbs.
1/8	0·052	2 1/8	15·1		
3/16	0·117	2 1/4	16·9	6	121·7
1/4	0·208	2 3/8	18·8	6 1/4	132·0
5/16	0·326	2 1/2	20·8	6 1/2	142·8
3/8	0·469	2 5/8	23·0	6 3/4	154·0
7/16	0·638	2 3/4	25·2		
1/2	0·833	2 7/8	27·6	7	165·6
				7 1/4	177·7
9/16	1·06	3	30·4	7 1/2	190·1
5/8	1·30	3 1/8	33·0	7 3/4	203·0
11/16	1·58	3 1/4	35·7		
3/4	1·88	3 3/8	38·5	8	216·3
13/16	2·20	3 1/2	41·4	8 1/4	230·1
7/8	2·55	3 5/8	44·4	8 1/2	244·2
15/16	2·93	3 3/4	47·5	8 3/4	258·8
		3 7/8	50·8		
1	3·33			9	273·8
1 1/16	3·76	4	54·1	9 1/4	289·2
1 1/8	4·22	4 1/8	57·5	9 1/2	305·1
1 3/16	4·70	4 1/4	61·1	9 3/4	321·3
1 1/4	5·21	4 3/8	64·7		
1 5/16	5·74	4 1/2	68·5	10	337·9
1 3/8	6·30	4 5/8	72·3	10 1/4	355·1
1 7/16	6·89	4 3/4	76·3	10 1/2	372·7
1 1/2	7·50	4 7/8	80·3	10 3/4	390·6
1 9/16	8·14				
1 5/8	8·80	5	84·5	11	408·9
1 11/16	9·60	5 1/8	88·8	11 1/4	427·8
1 3/4	10·2	5 1/4	93·2	11 1/2	447·0
1 13/16	11·0	5 3/8	97·7	11 3/4	466·7
1 7/8	11·7	5 1/2	102·2		
1 15/16	12·5	5 5/8	107·0	12	486·7
		5 3/4	111·8		
2	13·3	5 7/8	116·7		

TABLE 19.—WEIGHT OF FLAT WROUGHT-IRON BARS PER LINEAL FOOT.

Breadth of bar in inches	Thickness of bars in inches.														
	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1
	Weight of bars in pounds.														
$\frac{1}{2}$	·208	·312	·417	·521	·625	·729	·833	·937	1·04	1·15	1·25	1·35	1·46	1·56	1·67
$\frac{5}{8}$	·260	·391	·521	·651	·781	·911	1·04	1·17	1·30	1·43	1·56	1·69	1·82	1·95	2·08
$\frac{3}{4}$	·312	·469	·625	·781	·937	1·09	1·25	1·41	1·56	1·72	1·87	2·03	2·18	2·34	2·50
$\frac{7}{8}$	·365	·547	·729	·911	1·09	1·28	1·46	1·64	1·82	2·01	2·19	2·37	2·55	2·73	2·92
1	·417	·625	·833	1·04	1·25	1·46	1·67	1·88	2·08	2·29	2·50	2·71	2·92	3·13	3·33
$1\frac{1}{8}$	·469	·703	·938	1·17	1·41	1·64	1·88	2·11	2·34	2·58	2·81	3·05	3·28	3·52	3·75
$1\frac{1}{4}$	·521	·781	1·04	1·30	1·56	1·82	2·08	2·34	2·60	2·86	3·13	3·39	3·65	3·91	4·17
$1\frac{3}{8}$	·573	·859	1·15	1·43	1·72	2·01	2·29	2·58	2·86	3·15	3·44	3·72	4·01	4·30	4·58
$1\frac{1}{2}$	·625	·937	1·25	1·56	1·88	2·19	2·50	2·81	3·13	3·44	3·75	4·06	4·38	4·69	5·00
$1\frac{5}{8}$	·677	1·02	1·35	1·69	2·03	2·37	2·71	3·05	3·39	3·72	4·06	4·40	4·74	5·08	5·42
$1\frac{3}{4}$	·729	1·09	1·46	1·82	2·19	2·55	2·92	3·28	3·65	4·01	4·38	4·74	5·10	5·47	5·83
$1\frac{7}{8}$	·781	1·17	1·56	1·95	2·34	2·73	3·13	3·52	3·91	4·30	4·69	5·08	5·47	5·86	6·25
2	·833	1·25	1·67	2·08	2·50	2·92	3·33	3·75	4·17	4·58	5·00	5·42	5·83	6·25	6·67
$2\frac{1}{8}$		1·33	1·77	2·21	2·66	3·10	3·54	3·98	4·43	4·87	5·31	5·76	6·20	6·64	7·08
$2\frac{1}{4}$		1·41	1·88	2·34	2·81	3·28	3·75	4·22	4·69	5·16	5·63	6·09	6·56	7·03	7·50
$2\frac{3}{8}$		1·48	1·98	2·47	2·97	3·46	3·96	4·45	4·95	5·44	5·94	6·43	6·93	7·42	7·92
$2\frac{1}{2}$		1·56	2·08	2·60	3·13	3·65	4·17	4·69	5·21	5·73	6·25	6·77	7·29	7·81	8·33
$2\frac{5}{8}$		1·64	2·19	2·73	3·28	3·83	4·38	4·92	5·47	6·02	6·56	7·11	7·66	8·20	8·75
$2\frac{3}{4}$		1·72	2·29	2·86	3·44	4·01	4·58	5·16	5·73	6·30	6·88	7·45	8·02	8·59	9·17

TABLE 19 (*contd.*).—WEIGHT OF FLAT WROUGHT-IRON BARS PER LINEAL FOOT.

Breadth of bar in inches.	Thickness of bars in inches.														
	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1
	Weight of bars in pounds.														
$2\frac{7}{8}$		1·80	2·40	3·00	3·59	4·19	4·79	5·39	6·00	6·59	7·19	7·79	8·39	8·98	9·58
3		1·88	2·50	3·13	3·75	4·38	5·00	5·63	6·25	6·88	7·50	8·13	8·75	9·38	10·0
$3\frac{1}{4}$		2·03	2·71	3·39	4·06	4·74	5·42	6·09	6·77	7·45	8·13	8·80	9·48	10·2	10·8
$3\frac{1}{2}$			2·92	3·65	4·38	5·10	5·83	6·56	7·29	8·02	8·75	9·48	10·2	10·9	11·7
$3\frac{3}{4}$			3·13	3·91	4·69	5·47	6·25	7·03	7·81	8·59	9·38	10·2	10·9	11·7	12·5
4			3·33	4·17	5·00	5·83	6·67	7·50	8·33	9·17	10·0	10·8	11·7	12·5	13·3
$4\frac{1}{4}$			3·54	4·43	5·31	6·20	7·08	7·97	8·85	9·74	10·6	11·5	12·4	13·3	14·2
$4\frac{1}{2}$			3·75	4·69	5·63	6·56	7·50	8·44	9·38	10·3	11·3	12·2	13·1	14·1	15·0
$4\frac{3}{4}$			3·96	4·95	5·94	6·93	7·92	8·91	9·90	10·9	11·9	12·9	13·9	14·8	15·8
5			4·17	5·21	6·25	7·29	8·33	9·38	10·4	11·5	12·5	13·5	14·6	15·6	16·7
$5\frac{1}{2}$			4·58	5·73	6·88	8·02	9·17	10·3	11·5	12·6	13·8	14·9	16·0	17·2	18·3
6			5·00	6·25	7·50	8·75	10·0	11·3	12·5	13·8	15·0	16·3	17·5	18·8	20·0
$6\frac{1}{2}$			5·42	6·77	8·13	9·47	10·8	12·2	13·5	14·9	16·3	17·6	19·0	20·3	21·7
7			5·83	7·29	8·75	10·2	11·7	13·1	14·6	16·0	17·5	19·0	20·4	21·9	23·3
8			6·67	8·33	10·0	11·7	13·3	15·0	16·7	18·3	20·0	21·7	23·3	25·0	26·7
9			7·50	9·38	11·3	13·1	15·0	16·9	18·8	20·6	22·5	24·4	26·3	28·1	30·0
10			8·33	10·4	12·5	14·6	16·7	18·8	20·8	22·9	25·0	27·1	29·2	31·3	33·3
11			9·17	11·5	13·8	16·0	18·3	20·6	22·9	25·2	27·5	29·8	32·1	34·4	36·7
12			10·0	12·5	15·0	17·5	20·0	22·5	25·0	27·5	30·0	32·5	35·0	37·5	40·0

TABLE 20.—WEIGHT OF ROUND COPPER RODS.

Diameter in inches.	Wt. rods per foot length in lbs.
$\frac{1}{32}$	·003
$\frac{1}{16}$	·012
$\frac{3}{32}$	·027
$\frac{1}{8}$	·047
$\frac{5}{32}$	·074
$\frac{3}{16}$	·106
$\frac{7}{32}$	·144
$\frac{1}{4}$	·189
$\frac{3}{8}$	·425
$\frac{1}{2}$	·755
$\frac{5}{8}$	1·17
$\frac{3}{4}$	1·69
$\frac{7}{8}$	2·31
1	3·02
$1\frac{1}{8}$	3·82
$1\frac{1}{4}$	4·72
$1\frac{3}{8}$	5·72
$1\frac{1}{2}$	6·80
$1\frac{5}{8}$	7·98
$1\frac{3}{4}$	9·25
$1\frac{7}{8}$	10·6
2	12·1

TABLE 21.—WEIGHT OF SHEET LEAD.

Thickness in inches.	Wt. per square foot in lbs.
$\frac{1}{16}$	3·72
$\frac{1}{8}$	7·44
$\frac{3}{16}$	11·2
$\frac{1}{4}$	14·9
$\frac{5}{16}$	18·6
$\frac{3}{8}$	22·3
$\frac{7}{16}$	26·0
$\frac{1}{2}$	29·8
$\frac{9}{16}$	33·5
$\frac{5}{8}$	37·2
$\frac{11}{16}$	40·9
$\frac{3}{4}$	44·6
$\frac{7}{8}$	52·1
1	59·5
$1\frac{1}{8}$	66·9
$1\frac{1}{4}$	74·4
$1\frac{3}{8}$	81·8
$1\frac{1}{2}$	89·3
$1\frac{5}{8}$	96·7
$1\frac{3}{4}$	104
$1\frac{7}{8}$	112
2	119

TABLE 22.—WEIGHT OF BRASS.

Thickness of sheet or side of square or diameter.	Wt. sheets per sq. foot.	Wt. square bars per foot length.	Wt. rods per foot length.
Inches.	lbs.	lbs.	lbs.
1/32	1·36	·004	·003
1/16	2·71	·014	·011
3/32	4·06	·032	·025
1/8	5·42	·056	·044
5/32	6·75	·088	·069
3/16	8·13	·127	·100
7/32	9·50	·173	·136
1/4	10·8	·226	·177
5/16	13·5	·353	·277
3/8	16·3	·508	·399
7/16	19·0	·691	·543
1/2	21·7	·903	·709
9/16	24·3	1·14	·900
5/8	27·1	1·41	1·11
11/16	29·8	1·70	1·34
3/4	32·5	2·03	1·60
13/16	35·2	2·38	1·87
7/8	37·9	2·76	2·17
15/16	40·6	3·18	2·49
1	43·3	3·61	2·84
1 1/8	48·7	4·57	3·60
1 1/4	54·2	5·64	4·43
1 3/8	59·6	6·82	5·37
1 1/2	65·0	8·12	6·38
1 5/8	70·4	9·53	7·49
1 3/4	75·9	11·1	8·68
1 7/8	81·3	12·7	9·97
2	86·7	14·4	11·3

TABLE 23.—SECTIONAL AREA AND WEIGHT, ALUMINIUM BAR AND STRIP.

Dimensions.				Area.		Weight.	
Inches.		Millimetres.		Sq. ins.	Sq. mm.	Lbs./Ft.	Kg./M.
6	½	152	12·7	3·00	1940	3·52	5·25
6	⅜	152	9·52	2·25	1450	2·64	3·94
6	¼	152	6·35	1·50	968	1·76	2·62
5	½	127	12·7	2·50	1610	2·94	4·37
5	⅜	127	9·52	1·87	1210	2·20	3·28
5	¼	127	6·35	1·25	806	1·47	2·19
5	3/16	127	4·76	0·937	605	1·10	1·64
4½	½	114	12·7	2·25	1450	2·64	3·94
4½	⅜	114	9·52	1·69	1090	1·98	2·95
4½	¼	114	6·35	1·12	726	1·32	1·97
4½	3/16	114	4·76	0·844	544	0·991	1·47
4	½	102	12·7	2·00	1290	2·35	3·50
4	⅜	102	9·52	1·50	967	1·76	2·62
4	¼	102	6·35	1·00	645	1·17	1·75
3½	¼	88·9	6·35	0·875	564	1·03	1·53
3¼	½	82·5	12·7	1·62	1050	1·91	2·84
3¼	¼	82·5	6·35	0·812	524	0·955	1·42
3	¼	76·2	6·35	0·750	484	0·881	1·31
3	3/16	76·2	4·76	0·562	363	0·661	0·983
2¾	¼	69·4	6·35	0·687	444	0·808	1·20
2 9/16	⅜	65·1	9·52	0·961	619	1·13	1·68
2½	½	63·5	12·7	1·25	806	1·47	2·19
2½	5/16	63·5	7·94	0·781	504	0·918	1·37
2½	¼	63·5	6·35	0·625	403	0·734	1·09
2¼	¾	57·1	19·0	1·69	1090	1·98	2·95
2¼	½	57·1	12·7	1·12	726	1·32	1·97
2¼	⅜	57·1	9·52	0·844	544	0·991	1·47
2¼	5/16	57·1	7·94	0·703	454	0·826	1·23
2¼	¼	57·1	6·35	0·562	363	0·661	0·983
2¼	3/16	57·1	4·76	0·422	272	0·496	0·738
2	½	50·8	12·7	1·00	645	1·17	1·75
2	⅜	50·8	9·52	0·750	484	0·881	1·31
2	5/16	50·8	7·94	0·625	403	0·734	1·09
2	¼	50·8	6·35	0·500	323	0·587	0·875
2	3/16	50·8	4·76	0·375	242	0·441	0·656
2	⅛	50·8	3·17	0·250	161	0·294	0·437
1 13/16	5/16	46·0	7·94	0·566	365	0·665	0·990
1¾	⅝	44·4	15·9	1·09	706	1·28	1·91
1¾	5/16	44·4	7·94	0·547	353	0·642	0·956
1¾	¼	44·4	6·35	0·437	282	0·514	0·765

TABLE 23 (*contd.*).—SECTIONAL AREA AND WEIGHT, ALUMINIUM BAR AND STRIP.

Dimensions.				Area.		Weight.	
Inches.		Millimetres.		Sq. ins.	Sq. mm.	Lbs./Ft.	Kg./M.
$1\frac{3}{4}$	$\frac{1}{8}$	44·4	3·17	0·219	141	0·257	0·383
$1\frac{5}{8}$	$\frac{1}{4}$	41·3	6·35	0·406	262	0·477	0·710
$1\frac{5}{8}$	$\frac{3}{16}$	41·3	4·76	0·305	197	0·358	0·533
$1\frac{1}{2}$	$\frac{1}{2}$	38·1	12·7	0·750	484	0·881	1·31
$1\frac{1}{2}$	$\frac{3}{8}$	38·1	9·52	0·562	363	0·661	0·983
$1\frac{1}{2}$	$\frac{1}{4}$	38·1	6·35	0·375	242	0·441	0·656
$1\frac{1}{2}$	$\frac{3}{16}$	38·1	4·76	0·281	181	0·330	0·492
$1\frac{1}{2}$	$\frac{1}{8}$	38·1	3·17	0·187	121	0·220	0·328
$1\frac{7}{16}$	$\frac{1}{8}$	36·5	3·17	0·180	116	0·211	0·314
$1\frac{5}{16}$	$\frac{1}{8}$	33·3	3·17	0·164	106	0·193	0·287
$1\frac{1}{4}$	$\frac{3}{8}$	31·7	9·52	0·469	302	0·551	0·820
$1\frac{1}{4}$	$\frac{5}{16}$	31·7	7·94	0·391	252	0·459	0·683
$1\frac{1}{4}$	$\frac{1}{4}$	31·7	6·35	0·312	202	0·367	0·547
$1\frac{1}{4}$	$\frac{3}{16}$	31·7	4·76	0·234	151	0·275	0·410
$1\frac{1}{4}$	$\frac{1}{8}$	31·7	3·17	0·156	101	0·183	0·273
$1\frac{1}{4}$	$\frac{3}{32}$	31·7	2·38	0·117	75·6	0·138	0·205
$1\frac{1}{4}$	$\frac{1}{16}$	31·7	1·59	0·0781	50·4	0·0918	0·137
$1\frac{1}{8}$	$\frac{1}{2}$	28·6	12·7	0·562	363	0·661	0·983
$1\frac{1}{8}$	$\frac{1}{4}$	28·6	6·35	0·281	181	0·330	0·492
$1\frac{1}{8}$	$\frac{1}{8}$	28·6	3·17	0·141	90·7	0·165	0·246
1	$\frac{1}{2}$	25·4	12·7	0·500	323	0·587	0·875
1	$\frac{3}{8}$	25·4	9·52	0·375	242	0·441	0·656
1	$\frac{1}{4}$	25·4	6·35	0·250	161	0·294	0·437
1	$\frac{3}{16}$	25·4	4·76	0·187	121	0·220	0·328
1	$\frac{1}{8}$	25·4	3·17	0·125	80·6	0·147	0·219
1	$\frac{3}{32}$	25·4	2·38	0·0937	60·5	0·110	0·164
1	$\frac{}{16}$	25·4	1·59	0·0625	40·3	0·0734	0·109
$\frac{15}{16}$	$\frac{1}{16}$	23·8	1·59	0·0586	37·8	0·0688	0·102
$\frac{7}{8}$	$\frac{3}{8}$	22·2	9·52	0·328	212	0·385	0·574
$\frac{7}{8}$	$\frac{1}{4}$	22·2	6·35	0·219	141	0·257	0·383
$\frac{7}{8}$	$\frac{3}{16}$	22·2	4·76	0·164	106	0·193	0·287
$\frac{7}{8}$	$\frac{1}{8}$	22·2	3·17	0·109	70·6	0·128	0·191
$\frac{7}{8}$	·064	22·2	1·63	0·0560	36·1	0·0658	0·0979
$\frac{7}{8}$	·048	22·2	1·22	0·0420	27·1	0·0493	0·0734
$\frac{3}{4}$	$\frac{1}{4}$	19·0	6·35	0·187	121	0·220	0·328
$\frac{3}{4}$	$\frac{7}{32}$	19·0	5·56	0·164	106	0·193	0·287
$\frac{3}{4}$	$\frac{3}{16}$	19·0	4·76	0·141	90·7	0·165	0·246

TABLE 23 (*contd.*).—SECTIONAL AREA AND WEIGHT, ALUMINIUM BAR AND STRIP.

Dimensions.				Area.		Weight.	
Inches.		Millimetres.		Sq. ins.	Sq. mm.	Lbs./Ft.	Kg./M.
$\frac{3}{4}$	$\frac{1}{8}$	19·0	3·17	0·0937	60·5	0·110	0·164
$\frac{3}{4}$	·104	19·0	2·64	0·0780	50·3	0·0916	0·136
$\frac{3}{4}$	·064	19·0	1·63	0·0480	31·0	·0564	0·0839
$\frac{3}{4}$	·048	19·0	1·22	0·0360	23·2	0·0423	0·0629
$\frac{5}{8}$	$\frac{3}{16}$	15·9	4·76	0.117	75·6	0·138	0·205
$\frac{5}{8}$	$\frac{1}{8}$	15·9	3·17	0·0781	50·4	0·0918	0·137
$\frac{5}{8}$	·048	15·9	1·22	0·0300	19·4	0·0352	0·0525
$\frac{1}{2}$	$\frac{3}{16}$	12·7	4·76	0·0937	60·5	0·110	0·164
$\frac{1}{2}$	$\frac{1}{8}$	12·7	3·17	0·0625	40·3	0·0734	0·109
$\frac{1}{2}$	$\frac{3}{32}$	12·7	2·38	0·0469	30·2	0·0551	0·0820
$\frac{1}{2}$	·080	12·7	2·03	0·0400	25·8	0·0470	0·0699
$\frac{1}{2}$	·064	12·7	1·63	0·0320	20·6	0·0376	0·0560
$\frac{1}{2}$	·048	12·7	1·22	0·0240	15·5	0·0282	0·0420
$\frac{7}{16}$	$\frac{3}{32}$	11·1	2·38	0·0410	26·5	0·0482	0·0717
$\frac{7}{16}$	·064	11·1	1·63	0·0280	18·1	0·0329	0·0490
$\frac{7}{16}$	·048	11·1	1·22	0·0210	13·5	0·0247	0·0367
$\frac{3}{8}$	$\frac{3}{16}$	9·52	4·76	0·0703	45·4	0·0826	0·123
$\frac{3}{8}$	$\frac{1}{8}$	9·52	3·17	0·0469	30·2	0·0551	0·0820
$\frac{3}{8}$	·080	9·52	2·03	0·0300	19·4	0·0352	0·0525
$\frac{3}{8}$	·064	9·52	1·63	0·0240	15·5	0·0282	0·0420
$\frac{3}{8}$	·048	9·52	1·22	0·0180	11·6	0·0211	0·0315
$\frac{11}{32}$	·036	8·73	0·914	0·0124	7·98	0·0145	0·0216
$\frac{5}{16}$	·064	7·94	1·63	0·0200	12·9	0·0235	0·0350
$\frac{5}{16}$	·048	7·94	1·22	0·0150	9·68	0·0176	0·0262
$\frac{1}{4}$	·064	6·35	1·63	0·0160	10·3	0·0188	0·0280
$\frac{1}{4}$	$\frac{1}{16}$	6·35	1·59	0·0156	10·1	0·0183	0·0273
$\frac{1}{4}$	·048	6·35	1·22	0·0120	7·74	0·0141	0·0210
$\frac{1}{4}$	·036	6·35	0·914	0·00900	5·81	0·0106	0·0157
$\frac{3}{16}$	·064	4·76	1·63	0·0120	7·74	0·0141	0·0210
$\frac{3}{16}$	·048	4·76	1·22	0·00900	5·81	0·0106	0·0157
$\frac{1}{8}$	·036	3·17	0·914	0·00450	2·90	0·00529	0·00787

TABLE 24.—SECTIONAL AREA AND WEIGHT, ALUMINIUM WIRE.

S.W.G. Size.	Amps. at 600 per sq. in.	Diameter of Wire		Sect. Area of Wire.		Weight.	
		In.	Mm.	Sq. in.	Sq. mm.	Lbs. per 1000 yds.	Lbs. per mile.
7	14·6	0·176	4·47	0·0243	15·7	85·7	151
8	12·1	0·160	4·06	0·0201	13·0	70·9	125
9	9·77	0·144	3·66	0·0163	10·5	57·4	101
10	7·72	0·128	3·25	0·0129	8·30	45·4	79·8
11	6·34	0·116	2·95	0·0106	6·82	37·3	65·6
12	5·10	0·104	2·64	0·00849	5·48	29·9	52·7
13	3·99	0·092	2·34	0·00665	4·29	23·4	41·2
14	3·02	0·080	2·03	0·00503	3·24	17·7	31·2
15	2·44	0·072	1·83	0·00407	2·63	14·4	25·3
16	1·93	0·064	1·63	0·00322	2·07	11·3	20·0
17	1·48	0·056	1·42	0·00246	1·59	8·68	15·3
18	1·09	0·048	1·22	0·00181	1·17	6·38	11·2
19	0·754	0·040	1·02	0·00126	0·811	4·43	7·80
20	0·611	0·036	0·914	0·00102	0·657	3·59	6·31
21	0·482	0·032	0·813	0·000804	0·519	2·83	4·99
22	0·370	0·028	0·711	0·000616	0·397	2·17	3·82

TABLE 25.—WEIGHT AND SECTIONAL AREA, ALUMINIUM ROUND ROD.

Diameter of Rod.		Area.		Weight.	
Inches.	Mm.	Sq. ins.	Sq. mm.	Lbs./Ft.	Kg./M.
2	50·8	3·14	2030	3·69	5·49
$1\frac{31}{32}$	50·0	3·04	1960	3·58	5·32
$1\frac{15}{16}$	49·2	2·95	1900	3·46	5·15
$1\frac{13}{16}$	46·0	2·58	1660	3·03	4·51
$1\frac{3}{4}$	44·4	2·40	1550	2·83	4·21
$1\frac{23}{32}$	43·7	2·32	1500	2·73	4·06
$1\frac{11}{16}$	42·9	2·24	1440	2·63	3·91
$1\frac{5}{8}$	41·3	2·07	1340	2·44	3·63
$1\frac{9}{1}$	39·7	1·92	1240	2·25	3·35
$1\frac{1}{2}$	38·1	1·77	1140	2·08	3·09
$1\frac{7}{16}$	36·5	1·62	1050	1·91	2·84
$1\frac{13}{32}$	35·7	1·55	1000	1·82	2·72
$1\frac{5}{16}$	33·3	1·35	873	1·59	2·37
$1\frac{1}{4}$	31·7	1·23	792	1·44	2·15
$1\frac{3}{16}$	30·2	1·11	714	1·30	1·94
$1\frac{1}{8}$	28·6	0·994	641	1·17	1·74
1	25·4	0·785	507	0·923	1·37
$\frac{7}{8}$	22·2	0·601	388	0·706	1·05
$\frac{3}{4}$	19·0	0·442	285	0·519	0·772
$\frac{5}{8}$	15·9	0·307	198	0·360	0·536
$\frac{1}{2}$	12·7	0·196	127	0·231	0·343
$\frac{3}{8}$	9·52	0·110	71·3	0·130	0·193
$\frac{1}{4}$	6·34	0·0491	31·7	0·0577	0·0858

TABLE 26.—WEIGHT OF A SQUARE FOOT OF DIFFERENT METALS.

Thickness in fractions of an inch.

Thick-ness in inches.	Wrought-iron. lbs.	Cast-iron. lbs.	Steel. lbs.	Brass. lbs.	Copper. lbs.	Lead. lbs.	Zinc. lbs.
$\frac{1}{16}$	2·5	2·34	2·6	2·7	2·9	3·7	2·3
$\frac{1}{8}$	5·0	4·68	5·2	5·5	5·8	7·4	4·7
$\frac{3}{16}$	7·5	7·02	7·8	8·2	8·7	11·1	7·0
$\frac{1}{4}$	10·0	9·36	10·4	11·0	11·6	14·8	9·4
$\frac{5}{16}$	12·5	11·70	13·0	13·7	14·5	18·5	11·7
$\frac{3}{8}$	15·0	14·00	15·6	16·4	17·2	22·2	14·0
$\frac{7}{16}$	17·5	16·38	18·2	19·2	20·0	25·9	16·4
$\frac{1}{2}$	20·0	18·72	20·8	21·9	22·9	29·5	18·7
$\frac{9}{16}$	22·5	21·06	23·4	24·6	25·7	33·2	21·1
$\frac{5}{8}$	25·0	23·40	26·0	27·4	28·6	36·9	23·4
$\frac{11}{16}$	27·5	25·74	28·6	30·1	31·4	40·6	25·7
$\frac{3}{4}$	30·0	28·08	31·2	32·9	34·3	44·3	28·1
$\frac{13}{16}$	32·5	30·42	33·8	35·6	37·2	48·0	30·4
$\frac{7}{8}$	35·0	32·76	36·4	38·3	40·0	51·7	32·8
$\frac{15}{16}$	37·5	35·10	39·0	41·2	42·9	55·4	35·1
1	40·0	37·44	41·6	43·9	45·8	59·1	37·5

TABLE 27.—WEIGHT OF A SQUARE FOOT OF DIFFERENT METALS. *Thickness in Birmingham Wire Gauge.*

Thick-ness. B.W.G.	Sheet-iron. lbs.	Copper. lbs.	Brass. lbs.	Thick-ness. B.W.G.	Sheet-iron. lbs.	Copper. lbs.	Brass. lbs.
1	12·00	13·740	13·170	16	2·60	2·977	2·853
2	11·36	13·007	12·468	17	2·32	2·656	2·546
3	10·36	11·862	11·370	18	1·96	2·244	2·151
4	9·52	10·900	10·448	19	1·68	1·924	1·844
5	8·80	10·076	9·658	20	1·40	1·603	1·536
6	8·12	9·297	8·912	21	1·28	1·466	1·405
7	7·20	8·244	7·902	22	1·12	1·282	1·229
8	6·60	7·557	7·243	23	1·00	1·145	1·097
9	5·92	6·778	6·497	24	·88	1·008	·966
10	5·36	6·137	5·883	25	·80	·916	·878
11	4·80	5·496	5·268	26	·72	·824	·790
12	4·36	4·992	4·785	27	·64	·733	·702
13	3·80	4·351	4·170	28	·56	·641	·615
14	3·32	3·801	3·644	29	·52	·595	·579
15	2·88	3·298	3·161	30	·48	·550	·527

TABLE 28.—WEIGHT OF A SQUARE FOOT OF DIFFERENT METALS. *Thickness in Imperial Wire Gauge.*

Thickness. Gauge Number.	Thickness. Decimals of an inch.	Wrought-iron. lbs.	Steel. lbs.	Brass. lbs.	Gun-metal. lbs.	Copper. lbs.	Lead. lbs.	Zinc. lbs.	White-metal. lbs.
7/0	·500	19·87	20·40	21·86	22·68	22·87	29·59	18·72	20·88
6/0	·464	18·44	18·93	20·28	21·04	21·18	27·66	17·37	19·36
5/0	·432	17·16	17·62	18·87	19·58	19·75	25·74	16·19	18·02
4/0	·400	15·89	16·32	17·45	18·14	18·26	23·67	14·97	16·68
3/0	·372	14·78	15·18	16·28	16·86	17·01	22·17	13·94	15·52
2/0	·348	13·83	14·20	15·21	15·78	15·92	20·74	13·05	14·52
1/0	·324	12·87	13·22	14·16	14·68	14·81	19·31	12·14	13·51
1	·300	11·92	12·24	13·08	13·60	13·70	17·75	11·23	12·51
2	·278	10·97	11·26	12·07	12·52	12·63	16·45	10·35	11·51
3	·252	10·02	10·29	11·02	11·43	11·53	15·03	9·45	10·52
4	·232	9·22	9·47	10·14	10·57	10·61	13·83	8·70	9·68
5	·212	8·43	8·66	9·27	9·67	9·70	12·64	7·95	8·85
6	·192	7·63	7·84	8·40	8·75	8·78	11·44	7·20	8·01
7	·176	6·86	7·04	7·55	7·87	7·90	10·29	6·48	7·20
8	·160	6·36	6·53	7·00	7·29	7·32	9·54	6·00	6·68
9	·144	5·72	6·13	6·30	6·56	6·58	8·58	5·40	6·00
10	·128	5·08	5·22	5·58	5·83	5·85	7·62	4·80	5·33
11	·116	4·61	4·73	5·07	5·29	5·31	6·91	4·35	4·84
12	·104	4·13	4·24	4·54	4·73	4·75	6·20	3·89	4·34
13	·092	3·66	3·76	4·03	4·17	4·21	5·49	3·45	3·84
14	·080	3·18	3·26	3·50	3·63	3·66	4·77	3·00	3·34
15	·072	2·86	2·94	3·14	3·26	3·30	4·30	2·70	3·00
16	·064	2·54	2·60	2·80	2·89	2·92	3·81	2·40	2·66
17	·054	2·14	2·19	2·35	2·44	2·46	3·21	2·02	2·25
18	·048	1·91	1·96	2·10	2·18	2·20	2·86	1·80	2·00
19	·040	1·59	1·63	1·75	1·81	1·83	2·38	1·49	1·67
20	·036	1·43	1·47	1·57	1·63	1·64	2·14	1·35	1·50
21	·032	1·28	1·31	1·41	1·46	1·47	1·92	1·24	1·34
22	·028	1·11	1·14	1·22	1·26	1·28	1·66	1·05	1·16
23	·024	·95	·97	1·04	1·08	1·09	1·43	·89	1·00
24	·022	·87	·89	·96	·99	1·00	1·30	·82	·91
25	·020	·79	·81	·87	·90	·91	1·18	·74	·83
26	·018	·71	·73	·78	·81	·82	1·06	·67	·74
27	·0164	65	·67	·71	·74	·75	·97	·62	·68
28	·0148	·58	·60	·64	·65	·66	·87	·54	·61
29	·0136	·54	·55	·60	·61	·62	·81	·50	·57
30	·0124	·50	·51	·55	·57	·58	·75	·47	·52

TABLE 29.—WEIGHT OF 100 FEET OF WIRE OF DIFFERENT METALS. *Thickness in Imperial Wire Gauge.*

Size. Imperial Wire Gauge.	Iron. lbs.	Steel. lbs.	Brass. lbs.	Copper. lbs.
1	23·4	24·0	25·7	26·8
2	19·6	20·1	21·6	22·5
3	16·4	16·8	18·1	18·8
4	13·8	14·3	15·2	15·8
5	11·6	11·9	12·7	13·4
6	9·5	9·7	10·5	10·9
7	8·0	8·2	8·8	9·2
8	6·6	6·8	7·3	7·6
9	5·3	5·4	5·8	6·1
10	4·23	4·34	4·65	4·99
11	3·46	3·55	3·80	4·00
12	2·80	2·87	3·08	3·22
13	2·16	2·22	2·37	2·48
14	1·66	1·70	1·83	1·91
15	1·33	1·36	1·46	1·53
16	1·06	1·08	1·16	1·22
17	·80	·82	·88	·92
18	·60	·62	·66	·69
19	·40	·41	·44	·46
20	·33	·34	·36	·38

TABLE 30.—WEIGHT OF 100 FEET OF WIRE OF DIFFERENT METALS. *Thickness in Birmingham Wire Gauge.*

B.W.G.	Wrought-iron. lbs.	Steel. lbs.	Brass. lbs.	Copper. lbs.	B.W.G.	Wrought-iron. lbs	Steel. lbs.	Brass. lbs.	Copper. lbs.
0	30·58	30·92	33·43	35·17	11	4·13	4·18	4·52	4·75
1	25·75	26·04	28·15	29·62	12	3·14	3·18	3·43	3·61
2	21·34	21·57	23·32	24·54	13	2·34	2·36	2·55	2·69
3	18·02	18·22	19·70	20·72	14	1·69	1·71	1·85	1·95
4	15·11	15·28	16·52	17·38	15	1·37	1·39	1·50	1·58
5	12·46	12·59	13·62	14·33	16	1·05	1·06	1·15	1·21
6	11·45	11·57	12·51	13·16	17	·80	·81	·87	·92
7	9·25	9·35	10·11	10·64	18	·61	·62	·67	·70
8	7·29	7·37	7·97	8·38	19	·47	·48	·51	·54
9	6·60	6·68	7·22	7·59	20	·32	·33	·34	·37
10	4·96	5·02	5·43	5·71					

TABLE 31.—WEIGHT OF BLACK BOLTS AND NUTS.

Hexagon Head and Nut, Round Neck.

Length of bolt in inches from underside of head to point.	Diameter of bolt in inches.															
	1/4	5/16	3/8	7/16	1/2	9/16	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 3/4	2
	Weight of one bolt and nut in lbs.															
1	·031	·056	·092	·139	·200	·276	·369	·613								
1 1/8	·033	·059	·096	·144	·207	·285	·380	·628								
1 1/4	·035	·061	·100	·150	·214	·295	·391	·644	·989							
1 3/8	·036	·065	·103	·156	·221	·303	·403	·661	1·011							
1 1/2	·038	·068	·108	·161	·228	·312	·414	·677	1·033	1·489						
1 5/8	·040	·078	·112	·166	·236	·322	·425	·694	1·055	1·518						
1 3/4	·042	·073	·116	·172	·243	·330	·436	·709	1·077	1·546	2·118					
1 7/8	·044	·076	·120	·177	·250	·340	·448	·725	1·099	1·575	2·177					
2	·046	·078	·124	·183	·257	·349	·458	·742	1·121	1·604	2·214	2·951	3·851			
2 1/8	·048	·081	·128	·189	·264	·358	·470	·758	1·143	1·632	2·251	2·996	3·904			
2 1/4	·049	·085	·132	·194	·271	·367	·481	·773	1·165	1·661	2·286	3·041	3·959	5·032		
2 3/8	·051	·087	·136	·199	·279	·376	·492	·790	1·187	1·690	2·323	3·085	4·014	5·097		
2 1/2	·053	·090	·140	·205	·286	·385	·504	·806	1·209	1·718	2·359	3·131	4·067	5·161	7·904	11·473
2 3/4	·056	·095	·149	·216	·300	·404	·526	·838	1·252	1·776	2·432	3·220	4·176	5·290	8·078	11·702
3	·060	·101	·156	·227	·375	·422	·549	·871	1·296	1·833	2·468	3·309	4·285	5·420	8·255	11·932

3¼	·064	·107	·164	·238	·329	·439	·571	·903	1·340	1·891	2·577	3·398	4·393	5·549	8·431	12·162
3½	·067	·112	·173	·248	·343	·458	·592	·935	1·384	1·928	2·650	3·489	4·502	5·678	8·606	12·391
3¾	·071	·118	·180	·260	·358	·476	·616	·968	1·429	2·005	2·722	3·578	4·610	5·807	8·781	12·619
4	·074	·123	·189	·270	·372	·494	·638	1·000	1·473	2·062	2·795	3·667	4·718	5·936	8·957	12·849
4¼	·078	·129	·197	·282	·386	·513	·660	1·032	1·517	2·119	2·867	3·756	4·827	6·065	9·133	13·078
4½	·081	·135	·205	·292	·401	·531	·683	1·064	1·561	2·177	2·941	3·846	4·936	6·195	9·309	13·308
4¾	·085	·140	·213	·304	·415	·549	·705	1·097	1·605	2·234	3·013	3·936	5·044	6·323	9·484	13·537
5	·089	·145	·221	·315	·429	·567	·727	1·129	1·649	2·292	3·085	4·025	5·152	6·452	9·661	13·767
5¼	·092	·152	·229	·326	·444	·585	·750	1·161	1·692	2·348	3·158	4·114	5·261	6·581	9·836	13·996
5½	·096	·157	·237	·337	·458	·603	·772	1·193	1·736	2·406	3·230	4·204	5·369	6·710	10·011	14·226
5¾	·099	·162	·245	·348	·472	·621	·795	1·226	1·780	2·464	3·304	4·293	5·478	6·839	10·187	14·456
6	·103	·169	·254	·359	·487	·640	·817	1·258	1·824	2·521	3·376	4·383	5·587	6·968	10·363	14·685
6½	·110	·179	·269	·381	·515	·676	·863	1·323	1·913	2·635	3·521	4·562	5·804	7·227	10·714	15·144
7	·117	·191	·285	·403	·544	·712	·907	1·388	2·001	2·749	3·666	4·741	6·021	7·485	11·065	15·603
7½	·124	·202	·302	·425	·573	·749	·952	1·452	2·088	2·865	3·812	4·920	6·238	7·743	11·416	16·061
8		·213	·319	·447	·601	·785	·997	1·517	2·176	2·979	3·957	5·098	6·455	8·001	11·768	16·520
8½			·334	·469	·631	·822	1·041	1·581	2·264	3·094	4·102	5·278	6·671	8·259	12·120	16·980
9				·491	·659	·857	1·086	1·646	2·353	3·208	4·248	5·456	6·888	8·517	12·470	17·439
9½					·687	·876	1·131	1·710	2·441	3·323	4·393	5·636	7·106	8·775	12·822	17·898
10						·931	1·177	1·774	2·528	3·437	4·531	5·814	7·322	9·033	13·162	18·357
10½							1·221	1·839	2·616	3·553	4·684	5·993	7·529	9·292	13·535	18·816
11								1·904	2·704	3·667	4·824	6·172	7·757	9·550	13·876	19·274
11½									2·793	3·782	4·973	6·351	7·973	9·808	14·237	19·733
12										3·896	5·119	6·531	8·191	10·066	14·579	20·192

TABLE 32.—WEIGHT OF ANGLE IRONS AND TEE IRONS PER LINEAL FOOT.

Sum of the width and depth in inches.	Average thickness in fraction of an inch.												
	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	1
	Weight per lineal foot in pounds.												
2	·78	1·13	1·46	1·76	2·03								
2¼	·88	1·29	1·67	2·02	2·34								
2½	·99	1·45	1·88	2·28	2·66	3·01							
2¾	1·09	1·60	2·08	2·54	2·97	3·37							
3	1·20	1·76	2·29	2·80	3·28	3·74	4·17						
3¼	1·30	1·91	2·50	3·06	3·59	4·10	4·58						
3½	1·41	2·07	2·71	3·32	3·91	4·48	5·00						
3¾	1·51	2·23	2·92	3·58	4·22	4·84	5·42						
4		2·38	3·13	3·84	4·53	5·20	5·83	6·45					
4¼		2·54	3·33	4·10	4·84	5·56	6·25	6·91					
4½		2·70	3·54	4·36	5·16	5·92	6·67	7·38	8·07				
4¾		2·85	3·75	4·62	5·47	6·29	7·08	7·85	8·59				
5		3·01	3·96	4·88	5·78	6·65	7·50	8·32	9·11	10·03			
5¼		3·16	4·17	5·14	6·09	7·02	7·92	8·79	9·63	10·62			
5½		3·32	4·38	5·40	6·41	7·38	8·33	9·26	10·16	11·20			
5¾		3·48	4·58	5·66	6·72	7·75	8·75	9·73	10·68	11·78	12·50		
6			4·79	5·92	7·03	8·11	9·17	10·20	11·19	12·37	13·13	14·95	
6½			5·21	6·45	7·66	8·84	10·00	11·13	12·24	13·54	14·38	16·41	
7			5·63	6·97	8·28	9·57	10·83	12·07	13·28	14·70	15·63	17·86	

TABLE 32 (*contd.*).—WEIGHT OF ANGLE IRONS AND TEE IRONS PER LINEAL FOOT.

Sum of the width and depth in inches.	Average thickness in fraction of an inch.												
	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	1
	Weight per lineal foot in pounds.												
7½			6·04	7·49	8·91	10·30	11·67	13·01	14·32	15·87	16·88	19·32	21·67
8			6·46	8·01	9·53	11·03	12·50	13·94	15·36	17·03	18·13	20·78	23·33
8½			6·88	8·53	10·16	11·76	13·33	14·88	16·41	18·20	19·38	22·24	25·00
9			7·29	9·05	10·78	12·49	14·17	15·82	17·45	19·36	20·63	23·70	26·67
9½			7·71	9·57	11·41	13·22	15·00	16·76	18·49	20·53	21·88	25·16	28·33
10					12·03	13·95	15·83	17·70	19·53	21·69	23·13	26·61	30·00
10½					12·66	14·67	16·67	18·63	20·57	22·86	24·38	28·07	31·67
11					13·28	15·40	17·50	19·57	21·61	24·03	25·63	29·53	33·33
12					14·53	16·86	19·17	21·44	23·70	26·36	28·13	32·45	36·67
13						18·31	20·84	23·31	25·78	28·70	30·63	35·36	40·00
14						19·77	22·50	25·19	27·87	31·03	33·13	38·28	43·30
15						21·22	24·17	27·06	29·95	33·36	35·63	41·19	46·67
16						22·67	25·84	28·93	32·03	35·70	38·13	44·12	50·00

TABLE 33.—WEIGHT OF CAST-IRON BALLS.

Weight of ball in lbs.	Diameter of ball in inches and decimals.	Diameter of ball in inches and 64ths.	Weight of ball in lbs.	Diameter of ball in inches and decimals.	Diameter of ball in inches and 64ths.
1	1·9428	$1\frac{60}{64}$	41	6·6993	$6\frac{45}{64}$
2	2·4478	$2\frac{29}{64}$	42	6·7533	$6\frac{48}{64}$
3	2·8020	$2\frac{51}{64}$	43	6·8065	$6\frac{52}{64}$
4	3·0840	$3\frac{5}{64}$	44	6·8589	$6\frac{55}{64}$
5	3·3222	$3\frac{21}{64}$	45	6·9104	$6\frac{58}{64}$
6	3·5303	$3\frac{34}{64}$	46	6·9612	$6\frac{62}{64}$
7	3·7165	$3\frac{46}{64}$	47	7·0113	$7\frac{1}{64}$
8	3·8857	$3\frac{57}{64}$	48	7·0607	$7\frac{4}{64}$
9	4·0412	$4\frac{3}{64}$	49	7·1094	$7\frac{7}{64}$
10	4·1857	$4\frac{12}{64}$	50	7·1574	$7\frac{10}{64}$
11	4·3208	$4\frac{21}{64}$	51	7·2048	$7\frac{13}{64}$
12	4·4480	$4\frac{29}{64}$	52	7·2516	$7\frac{16}{64}$
13	4·5682	$4\frac{36}{64}$	53	7·2978	$7\frac{19}{64}$
14	4·6825	$4\frac{44}{64}$	54	7·3434	$7\frac{22}{64}$
15	4·7914	$4\frac{51}{64}$	55	7·3885	$7\frac{25}{64}$
16	4·8956	$4\frac{57}{64}$	56	7·4330	$7\frac{28}{64}$
17	4·9956	5	57	7·4770	$7\frac{31}{64}$
18	5·0916	$5\frac{6}{64}$	58	7·5204	$7\frac{33}{64}$
19	5·1842	$5\frac{12}{64}$	59	7·5634	$7\frac{36}{64}$
20	5·2736	$5\frac{18}{64}$	60	7·6059	$7\frac{39}{64}$
21	5·3601	$5\frac{23}{64}$	61	7·6479	$7\frac{41}{64}$
22	5·4439	$5\frac{28}{64}$	62	7·6895	$7\frac{44}{64}$
23	5·5251	$5\frac{34}{64}$	63	7·7306	$7\frac{47}{64}$
24	5·6041	$5\frac{39}{64}$	64	7·7713	$7\frac{49}{64}$
25	5·6809	$5\frac{44}{64}$	65	7·8116	$7\frac{52}{64}$
26	5·7556	$5\frac{48}{64}$	66	7·8514	$7\frac{54}{64}$
27	5·8285	$5\frac{53}{64}$	67	7·8909	$7\frac{57}{64}$
28	5·8996	$5\frac{58}{64}$	68	7·9299	$7\frac{60}{64}$
29	5·9690	$5\frac{62}{64}$	69	7·9686	$7\frac{62}{64}$
30	6·0368	$6\frac{2}{64}$	70	8·0069	8
31	6·1032	$6\frac{7}{64}$	71	8·0449	$8\frac{3}{64}$
32	6·1681	$6\frac{11}{64}$	72	8·0825	$8\frac{5}{64}$
33	6·2317	$6\frac{15}{64}$	73	8·1197	$8\frac{8}{64}$
34	6·2940	$6\frac{19}{64}$	74	8·1566	$8\frac{10}{64}$
35	6·3551	$6\frac{23}{64}$	75	8·1932	$8\frac{12}{64}$
36	6·4151	$6\frac{27}{64}$	76	8·2295	$8\frac{15}{64}$
37	6·4740	$6\frac{30}{64}$	77	8·2654	$8\frac{17}{64}$
38	6·5317	$6\frac{34}{64}$	78	8·3010	$8\frac{19}{64}$
39	6·5885	$6\frac{38}{64}$	79	8·3364	$8\frac{22}{64}$
40	6·6444	$6\frac{41}{64}$	80	8·3714	$8\frac{24}{64}$

TABLE 33 (*contd.*).—WEIGHT OF CAST-IRON BALLS.

Weight of ball in lbs.	Diameter of ball in inches and decimals.	Diameter of ball in inches and 64ths.	Weight of ball in lbs.	Diameter of ball in inches and decimals.	Diameter of ball in inches and 64ths.
81	8·4061	$8\frac{26}{64}$	97	8·9267	$8\frac{59}{64}$
82	8·4406	$8\frac{28}{64}$	98	8·9573	$8\frac{61}{64}$
83	8·4747	$8\frac{30}{64}$	99	8·9876	$8\frac{63}{64}$
84	8·5086	$8\frac{33}{64}$	100	9·0178	$9\frac{1}{64}$
85	8·5423	$8\frac{35}{64}$	101	9·0478	$9\frac{3}{64}$
86	8·5756	$8\frac{37}{64}$	102	9·0775	$9\frac{5}{64}$
87	8·6088	$8\frac{39}{64}$	103	9·1071	$9\frac{7}{64}$
88	8·6416	$8\frac{41}{64}$	104	9·1365	$9\frac{9}{64}$
89	8·6742	$8\frac{43}{64}$	105	9·1657	$9\frac{11}{64}$
90	8·7066	$8\frac{45}{64}$	106	9·1947	$9\frac{12}{64}$
91	8·7387	$8\frac{47}{64}$	107	9·2235	$9\frac{14}{64}$
92	8·7706	$8\frac{49}{64}$	108	9·2521	$9\frac{16}{64}$
93	8·8023	$8\frac{51}{64}$	109	9·2806	$9\frac{18}{64}$
94	8·8337	$8\frac{53}{64}$	110	9·3089	$9\frac{20}{64}$
95	8·8649	$8\frac{55}{64}$	111	9·3370	$9\frac{22}{64}$
96	8·8959	$8\frac{57}{64}$	112	9·3650	$9\frac{23}{64}$

TABLE 34.—WEIGHT OF CAST-IRON PIPES AND CYLINDERS.

Bore of pipe in inches.	Thickness of metal in fractions of an inch.												
	¼	5/16	⅜	7/16	½	9/16	⅝	11/16	¾	⅞	1	1⅛	1¼
	Weight of cast-iron flange pipes per foot run in lbs.												
1	3·1	4·0	5·1	6·2	7·4	8·6	10·0	11·4	12·9	16·1	19·6	23·4	27·6
1¼	3·7	4·8	6·0	7·2	8·6	10·0	11·5	13·1	14·7	18·2	22·1	26·2	30·7
1½	4·3	5·6	6·9	8·3	9·8	11·4	13·0	14·7	16·5	20·4	24·5	28·9	33·7
1¾	4·9	6·3	7·8	9·4	11·0	12·7	14·5	16·4	18·4	22·5	26·9	31·7	36·7
2			8·7	10·4	12·3	14·1	16·1	18·1	20·3	24·7	29·5	34·5	39·9
2¼			9·6	11·5	13·5	15·5	17·6	19·8	22·0	26·8	31·8	37·2	42·9
2½			10·6	12·6	14·7	16·9	19·1	21·5	23·9	28·9	34·3	40·0	45·9
2¾			11·5	13·7	16·0	18·3	20·7	23·2	25·7	31·1	36·8	42·7	49·0
3			12·4	14·7	17·1	19·6	22·2	24·8	27·6	33·3	39·3	45·5	52·0
3¼			13·3	15·8	18·4	21·0	23·7	26·5	29·4	35·4	41·6	48·2	55·1
3½			14·2	16·9	19·6	22·4	25·3	28·2	31·2	37·5	44·1	51·0	58·2
3¾			15·2	18·0	20·8	23·8	26·8	29·9	33·1	39·7	46·6	53·8	61·3
4			16·1	19·0	22·1	25·1	28·3	31·6	35·0	41·9	49·1	56·5	64·3
4¼			17·0	20·1	23·3	26·6	30·0	33·3	36·8	44·0	51·5	59·3	67·4
4½			17·9	21·2	24·5	27·9	31·4	34·9	38·6	46·1	53·9	62·0	70·4
4¾			18·8	22·3	25·7	29·3	33·0	36·6	40·4	48·2	56·3	64·8	73·5
5			19·8	23·3	26·9	30·7	34·4	38·3	42·3	50·5	58·8	65·7	76·6
5¼			20·7	24·4	28·2	32·0	36·0	40·0	44·1	52·5	61·3	70·3	79·7
5½			21·6	25·5	29·4	33·5	37·5	41·7	46·0	54·7	63·7	73·1	82·7
5¾			22·5	26·6	30·6	34·8	39·1	43·4	47·8	56·8	66·2	75·8	85·8
6			23·4	27·6	31·9	36·2	40·6	45·0	49·7	59·1	68·7	78·7	89·0
6¼			24·4	28·7	33·1	37·6	42·9	46·8	51·5	61·1	71·1	81·3	92·0
6½			25·3	29·7	34·3	38·9	44·2	48·4	53·3	63·2	73·5	84·1	95·0
6¾			26·2	30·8	35·5	40·3	45·2	50·1	55·1	65·4	76·0	86·8	98·0
7			27·1	31·9	36·8	41·7	46·7	51·8	56·8	67·7	78·5	89·7	101·2
7¼			28·0	32·9	37·7	43·1	48·2	53·5	58·8	69·7	80·9	92·4	104·1
7½			29·0	34·0	39·2	44·5	49·8	55·2	60·7	71·9	83·3	95·1	107·2
7¾			29·9	35·1	40·4	45·8	51·3	56·8	62·5	74·0	85·8	97·9	110·3

8			30·8	36·2	41·6	47·1	52·8	58·5	64·3	76·2	88·4	100·3	113·5
8½			32·8	38·2	44·1	50·0	56·0	61·9	68·0	80·4	93·1	106·1	119·5
9			34·4	40·5	46·6	52·7	58·9	65·3	71·7	84·8	98·2	111·8	125·8
9½			36·3	42·6	49·0	55·5	62·0	68·6	75·3	89·1	102·7	117·1	131·4
10			38·2	44·7	52·4	58·2	65·1	72·9	79·0	93·3	108·0	122·9	138·1
10½			40·0	46·8	53·7	60·9	68·0	75·2	82·4	97·4	112·7	128·3	143·9
11			41·9	48·9	56·4	63·6	71·0	78·5	86·4	101·8	117·8	133·9	150·3
11½			43·7	51·2	59·0	66·5	74·3	82·1	90·2	106·1	122·6	139·2	156·3
12			45·6	53·4	61·4	69·3	77·3	85·8	93·7	110·4	127·4	145·0	162·6
12½					63·6	72·1	80·2	88·9	97·3	114·5	132·3	150·2	168·5
13					66·1	74·6	83·3	92·0	101·1	118·9	137·3	155·7	174·4
13½					68·6	77·4	86·4	95·5	104·8	123·2	142·0	161·3	180·6
14					71·2	80·2	89·6	98·9	108·4	127·5	147·0	167·1	187·2
14½					73·3	83·0	92·7	102·3	111·2	131·7	149·9	172·2	192·8
15					76·1	85·8	95·9	105·5	115·7	136·1	156·8	177·7	199·4
15½					78·6	88·6	99·0	109·2	119·5	140·7	161·9	183·5	205·3
16					81·0	91·4	102·0	112·6	123·2	144·7	166·6	188·7	211·7
16½					83·3	93·9	104·8	115·8	126·7	148·8	171·6	194·4	217·5
17					85·8	97·0	108·0	119·2	130·7	153·5	176·6	199·0	223·7
17½					88·3	99·5	111·1	122·6	134·2	157·6	181·3	205·3	229·6
18					90·8	102·3	114·3	126·0	137·9	161·9	186·2	210·9	236·2
18½					93·0	104·8	117·0	129·2	141·3	166·0	190·9	216·2	241·6
19					95·5	108·0	120·1	132·6	145·1	170·4	195·9	221·8	248·0
19½					98·0	110·8	123·2	136·0	148·8	174·7	200·9	227·4	255·5
20					100·5	113·3	126·5	139·5	152·3	178·9	205·8	232·9	260·3
20½							129·5	142·9	156·3	183·5	210·9	238·4	266·4
21							132·6	146·3	160·1	187·5	215·6	244·0	272·6
21½							135·4	149·4	163·5	191·9	220·6	249·6	278·6
22							138·8	152·9	167·2	197·1	225·4	254·9	284·8
22½							141·6	156·0	170·7	200·3	230·3	260·5	290·8
23							144·8	159·7	174·7	205·0	235·3	266·1	297·3
23½							147·9	162·9	178·2	209·0	240·2	271·4	303·3
24							151·2	166·6	182·5	213·9	245·0	276·9	309·3

The weight of the two flanges is usually reckoned equal to the weight of one foot of pipe.

TABLE 35.—WEIGHT OF SOLID DRAWN WROUGHT-IRON PIPES *per foot run; internal diameter.*

Internal diameter of pipe in inches.	Thickness of metal in parts of an inch.							
	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$
	Weight per foot run in pounds.							
$\frac{1}{4}$	·21	·50	·9	1·3	1·9	2·5	3·1	3·9
$\frac{3}{8}$	·30	·70	1·1	1·6	2·3	2·9	3·8	4·6
$\frac{1}{2}$	·40	·83	1·4	2·0	2·7	3·5	4·3	5·3
$\frac{5}{8}$	·46	1·1	1·6	2·3	3·1	3·9	4·9	5·9
$\frac{3}{4}$	·54	1·2	1·9	2·6	3·5	4·5	5·5	6·6
$\frac{7}{8}$	·60	1·3	2·1	2·9	3·9	4·9	6·1	7·3
1	·70	1·5	2·4	3·3	4·3	5·5	6·7	7·9
$1\frac{1}{4}$	·87	1·8	2·9	3·9	5·2	6·4	7·8	9·3
$1\frac{1}{2}$	1·00	2·1	3·3	4·7	5·9	7·4	8·9	10·6
$1\frac{3}{4}$	1·20	2·5	3·8	5·3	6·8	8·4	10·1	11·9
2	1·40	2·8	4·3	5·9	7·6	9·5	11·3	13·2
$2\frac{1}{4}$	1·53	3·1	4·8	6·6	8·5	10·4	12·4	14·5
$2\frac{1}{2}$	1·70	3·5	5·3	7·3	9·3	11·4	13·6	15·9
$2\frac{3}{4}$	1·90	3·8	5·8	7·9	10·1	12·4	14·7	17·2
3	2·03	4·1	6·3	8·6	10·9	13·3	15·9	18·5

TABLE 36.—WEIGHT OF SOLID DRAWN WROUGHT-IRON PIPES *per foot run; external diameter.*

External diameter in inches.	Thickness in parts of an inch.		
	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$
	Weight per foot in pounds.		
2	2·47	3·58	4·61
$2\frac{1}{8}$	2·63	3·88	4·94
$2\frac{1}{4}$	2·79	4·07	5·27
$2\frac{3}{8}$	2·96	4·32	5·59
$2\frac{1}{2}$	3·14	4·57	5·92
$2\frac{5}{8}$	3·29	4·81	6·25
$2\frac{3}{4}$	3·46	5·06	6·58
$2\frac{7}{8}$	3·61	5·30	6·91
3	3·78	5·55	7·24
$3\frac{1}{8}$	3·95	5·80	7·57
$3\frac{1}{4}$	4·12	6·05	7·90

TABLE 37.—WEIGHT OF LARGE WELDED WROUGHT-IRON PIPES *per foot run.*

Internal diameter of pipe in inches.	Thickness in parts of an inch.			
	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$
	Weight per foot in pounds.			
10	20	26	33	40
11	22	29	37	44
12	24	32	40	48
13	26	34	43	52
14	28	37	47	56
15	30	40	50	60
16	32	42	53	64
17	33	45	56	68
18	35	47	59	72
19	37	50	63	76
20	39	53	66	80
21	41	55	69	84
22	43	58	73	88
23	45	61	76	92
24	47	63	79	96

TABLE 38.—WEIGHT OF RIVETED WROUGHT-IRON PIPES *per foot run.*

Internal diameter of pipe in inches.	Thickness of pipe in parts of an inch.				
	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$
	Weight per foot in pounds.				
12	19	29	38	48	58
15	23	34	46	59	71
18	28	41	55	70	84
24	36	54	72	91	109
30	44	66	89	112	134
36	53	79	105	133	159
42	61	92	122	153	184
48	70	104	139	161	209

TABLE 39.—WEIGHT OF ANGLE-IRON FLANGE RINGS FOR RIVETED IRON PIPES.

Bore of pipe in inches.	Size of angle-iron.			
	$2 \times 2 \times \frac{3}{8}$	$2\frac{1}{2} \times 2\frac{1}{2} \times \frac{3}{8}$	$3 \times 3 \times \frac{1}{2}$	$3\frac{1}{2} \times 3\frac{1}{2} \times \frac{5}{8}$
	Weight of each ring in pounds.			
10	15	20	36	51
11	16	22	39	54
12	18	23	41	58
13	19	25	44	61
14	20	26	47	65
15	21	28	49	68
16	22	29	52	72
17	24	31	55	75
18	25	32	57	79
19	26	34	60	82
20	27	35	62	86
21	28	37	65	89
22	30	38	68	93
23	31	40	70	96
24	32	41	73	100

TABLE 40.—WEIGHT OF STEEL SOCKETS RIVETED ON PIPES.

Bore of pipe in inches.	Weight of socket in lbs.	Bore of pipe in inches.	Weight of socket in lbs.
10	10	18	25
11	11	19	29
12	12	20	32
13	13	21	34
14	14	22	35
15	16	23	37
16	19	24	38
17	22		

TABLE 41.—WEIGHT OF STEEL TUBES IN LBS. *per foot run* (Accles and Pollock).

Thickness of walls in S.W.G.	Outside diameter in inches.													
	⅜	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1⅞	2
10	·342	·519	·697	·874	1·051	1·229	1·406	1·549	1·723	1·896	2·069	2·239	2·409	2·587
11	·324	·483	·642	·800	·959	1·118	1·277	1·436	1·594	1·753	1·912	2·071	2·230	2·389
12	·307	·451	·595	·740	·884	1·028	1·172	1·317	1·461	1·605	1·749	1·894	2·038	2·182
13	·282	·407	·533	·659	·785	·910	1·036	1·162	1·287	1·413	1·539	1·664	1·790	1·916
14	·257	·367	·476	·586	·696	·806	·916	1·025	1·135	1·245	1·355	1·465	1·575	1·685
15	·231	·326	·422	·517	·612	·707	·803	·898	·993	1·088	1·184	1·279	1·375	1·470
16	·213	·300	·385	·471	·557	·643	·729	·815	·901	·987	1·073	1·159	1·245	1·331
17	·195	·272	·348	·425	·501	·578	·655	·732	·808	·885	·962	1·039	1·115	1·192
18	·169	·234	·299	·364	·428	·493	·558	·623	·688	·747	·808	·872	·936	1·000
19	·148	·204	·259	·315	·370	·426	·482	·537	·570	·624	·677	·720	·784	·837
20	·126	·172	·219	·265	·311	·358	·404	·450	·515	·563	·644	·659	·707	·755
21	·116	·159	·201	·243	·285	·328	·370	·413	·459	·492	·544	·587	·630	·672
22	·103	·140	·177	·214	·251	·288	·325	·362	·403	·440	·477	·515	·552	·589

TABLE 42.—WEIGHT OF BRASS TUBES IN LBS. *per foot run.*

Thickness of walls in S.W.G.	Outside diameter in inches.													
	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2
10	·3612	·548	·736	·923	1·11	1·298	1·483	1·634	1·820	2·005	2·184	2·361	2·542	2·733
11	·3421	·510	·6775	·845	1·011	1·18	1·348	1·515	1·682	1·852	2·02	2·184	2·355	2·522
12	·3242	·476	·628	·781	·934	1·084	1·238	1·390	1·542	1·695	1·846	2·0	2·150	2·303
13	·2979	·430	·563	·696	·829	·9615	1·093	1·228	1·359	1·492	1·623	1·757	1·891	2·021
14	·2714	·3875	·5025	·619	·735	·851	·968	1·072	1·199	1·314	1·43	1·547	1·661	1·779
15	·244	·344	·445	·546	·646	·746	·848	·948	1·048	1·148	1·25	1·35	1·451	1·552
16	·225	·3167	·4064	·497	·588	·679	·77	·861	·952	1·04	1·132	1·223	1·314	1·405
17	·206	·2871	·3675	·4487	·529	·6102	·692	·773	·854	·935	1·015	1·097	1·177	1·260
18	·1785	·247	·3158	·3845	·452	·5205	·589	·658	·7265	·788	·854	·921	·989	1·056
19	·1563	·2155	·2734	·3325	·3906	·450	·509	·567	·6015	·659	·715	·761	·828	·884
20	·133	·1816	·2313	·280	·3285	·378	·4265	·4755	·544	·5945	·680	·695	·746	·7975
21	·1224	·1679	·2122	·2566	·3008	·3463	·3905	·436	·485	·5195	·575	·620	·6655	·7098
22	·1087	·1478	·1869	·226	·2651	3 041	·343	·3823	·426	·4645	·504	·544	·5828	·622

TABLE 43.—WEIGHT OF BRASS TUBES *per foot run.*
Thickness in parts of an inch.

Diameter of bore of tube in inches.	Thickness in parts of an inch.		
	⅛	3/16	¼
	Weight per foot run in lbs. and ounces.		
1	1 9½	2 4	3 8½
1⅛	1 12¼	2 8½	3 14½
1¼	1 15½	2 12¼	4 4
1⅜	2 2	3 0½	4 10
1½	2 5	3 4½	4 15½
1⅝	2 7½	3 8½	5 5
1¾	2 10½	3 12½	5 11
1⅞	2 13½	4 0½	6 0½
2	3 0	4 5	6 6½
2⅛	3 3	4 9¼	6 12
2¼	3 6	4 13¼	7 2
2⅜	3 9	5 1½	7 7¼
2½	3 11¾	5 6	7 12
2⅝	3 14½	5 10	8 2½
2¾	4 1½	5 14	8 8½
2⅞	4 4¼	6 2	8 14
3	4 7	6 6	9 4
3¼	4 12½	6 14½	9 15½
3½	5 2¼	7 6½	10 11
3¾	5 8	7 15	11 5¾
4	5 14	8 7½	12 1
5	7 4	10 8	14 14½
6	8 11	12 10	17 12

TABLE 44.—WEIGHT OF SOLID-DRAWN COPPER TUBES *per foot run.*

Diameter of bore of tube inches.	Imperial wire gauge.												
	6	7	8	9	10	11	12	13	14	15	16	17	18
	Thickness in Millimetres.												
	·192	·176	·160	·144	·128	·116	·104	·092	·080	·072	·064	·056	·048
	Weight per lineal foot in pounds.												
3/8	1·32	1·17	1·04	0·90	0·78	0·70	0·60	0·52	0·44	0·39	0·34	0·30	0·25
1/2	1·61	1·44	1·28	1·12	0·97	0·87	0·76	0·66	0·56	0·50	0·44	0·38	0·32
5/8	1·90	1·70	1·52	1·34	1·17	1·05	0·92	0·80	0·68	0·61	0·54	0·47	0·39
3/4	2·19	1·97	1·77	1·56	1·37	1·22	1·08	0·94	0·80	0·72	0·63	0·55	0·47
1	2·79	2·51	2·26	2·00	1·76	1·58	1·39	1·21	1·04	0·93	0·83	0·72	0·61
1¼	3·36	3·05	2·74	2·44	2·15	1·93	1·71	1·49	1·29	1·15	1·02	0·89	0·76
1½	3·98	3·60	3·23	2·88	2·54	2·28	2·03	1·77	1·53	1·37	1·22	1·06	0·91
1¾	4·53	4·12	3·72	3·31	2·93	2·63	2·34	2·05	1·77	1·59	1·41	1·23	1·05
2	5·11	4·66	4·20	3·75	3·32	2·99	2·66	2·33	2·02	1·81	1·61	1·40	1·20
2½	6·28	5·73	5·18	4·63	4·10	3·70	3·30	2·89	2·50	2·25	2·00	1·76	1·49
3	7·46	6·80	6·15	5·50	4·87	4·40	3·93	3·45	3·00	2·68	2·38	2·08	1·77
3½	8·62	7·87	7·12	6·39	5·66	5·11	4·56	4·01	3·48	3·12	2·77	2·41	2·06
4	9·78	8·94	8·10	7·26	6·43	5·81	5·20	4·57	3·97	3·56	3·17	2·75	2·35
4½	10·95	10·00	9·05	8·13	7·21	6·53	5·83	5·13	4·45	4·00	3·53	3·09	2·64
5	12·13	11·08	10·05	9·00	8·00	7·23	6·46	5·69	4·94	4·44	3·92	3·42	2·93
5½	13·30	12·16	11·02	9·90	8·78	7·94	7·10	6·25	5·43	4·85	4·31	3·76	3·22
6	14·50	13·25	12·00	10·75	9·56	8·66	7·74	6·81	5·91	5·29	4·69	4·10	3·51

TABLE 45.—WEIGHT OF COPPER TUBES *per foot run*.

Diameter of bore of tube, inches.	Thickness in parts of an inch.					
	⅛		$\frac{3}{16}$		¼	
	Weight per foot run in lbs. and ounces.					
½	0	15	1	10¼	2	4¼
⅝	1	2½	1	14	2	10½
¾	1	5½	2	3	3	0½
⅞	1	8¼	2	7½	3	6½
1	1	11	2	11½	3	12½
1⅛	1	14¼	3	0	4	2
1¼	2	1¼	3	4¾	4	8
1⅜	2	3¾	3	8¾	4	14
1½	2	7	3	13½	5	4
1⅝	2	11	4	2½	5	10
1¾	2	14	4	7	6	0
1⅞	3	0½	4	11¼	6	6
2	3	3½	4	15¾	6	12
2⅛	3	6¼	5	4	7	2
2¼	3	9½	5	8½	7	8
2⅜	3	12	5	13¼	7	14
2½	3	15½	6	1¾	8	4
2⅝	4	2¼	6	6	8	10
2¾	4	5½	6	10¼	9	0
2⅞	4	9	6	15½	9	6
3	4	12½	7	4¼	9	12
3⅛	4	15	7	9	10	4

Table 46.—Aluminium Tube.

Weight in Lbs. per Lineal Foot.

S. W. G. No.		1	2	3	4	5	6	7	8	9
Thickness	Ins	0·300	0·276	0·252	0·232	0·212	0·192	0·176	0·160	0·144
,,	Mm	7·62	7·00	6·40	5·89	5·38	4·87	4·46	4·06	3·66
Outside Dia. Ins.	Mm.									
3/16	4·76									
¼	6·35									
⅜	9·52									
½	12·7									0·189
⅝	15·9							0·292	0·275	0·259
¾	19·0					0·420	0·394	0·373	0·349	0·321
⅞	22·2			0·580	0·550	0·520	0·484	0·454	0·423	0·388
1	25·4	0·775	0·740	0·700	0·656	0·615	0·575	0·535	0·497	0·455
1⅛	28·6	0·915	0·872	0·810	0·761	0·715	0·663	0·616	0·570	0·521
1¼	31·8	1·05	0·990	0·926	0·870	0·813	0·751	0·699	0·646	0·588
1⅜	34·9	1·18	1·18	1·04	0·980	0·910	0·840	0·776	0·716	0·653
1½	38·1	1·33	1·25	1·16	1·09	1·01	0·931	0·862	0·793	0·722
1⅝	41·3	1·47	1·37	1·27	1·19	1·10	1·02	0·943	0·865	0·788
1¾	44·5	1·61	1·50	1·39	1·30	1·21	1·10	1·02	0·940	0·852
1⅞	47·6	1·75	1·63	1·51	1·41	1·30	1·19	1·10	1·01	0·920
2	50·8	1·89	1·76	1·63	1·51	1·40	1·28	1·19	1·09	0·986
2⅛	53·9	2·03	1·89	1·73	1·62	1·50	1·37	1·27	1·16	1·06
2¼	57·1	2·16	2·01	1·86	1·73	1·60	1·46	1·35	1·23	1·12
2⅜	60·3	2·30	2·14	1·98	1·84	1·69	1·55	1·43	1·31	1·19
2½	63·5	2·44	2·27	2·10	1·95	1·80	1·64	1·51	1·39	1·25
2⅝	66·6	2·58	2·40	2·21	2·05	1·89	1·72	1·59	1·46	1·32
2¾	69·8	2·72	2·53	2·35	2·17	2·00	1·82	1·67	1·53	1·39
2⅞	73·0	2·86	2·65	2·45	2·27	2·09	1·90	1·75	1·61	1·45
3	76·2	2·99	2·78	2·56	2·37	2·18	1·99	1·84	1·68	1·51
3¼	82·6	3·27	3·03	2·79	2·58	2·38	2·16	2·00	1·83	1·65
3½	88·9	3·55	3·29	3·03	2·81	2·58	2·35	2·16	1·96	1·78
3¾	95·2	3·83	3·64	3·25	3·01	2·77	2·53	2·33	2·12	1·91
4	102	4·10	3·80	3·49	3·23	2·97	2·70	2·49	2·27	2·05
4¼	108	4·38	4·05	3·72	3·45	3·16	2·88	2·65	2·42	2·19
4½	114	4·66	4·31	3·96	3·66	3·36	3·06	2·82	2·57	2·32
4¾	121	4·94	4·57	4·19	3·88	3·56	3·24	2·98	2·72	2·45
5	127	5·21	4·82	4·42	4·09	3·76	3·41	3·14	2·86	2·58
5¼	133	5·49	5·07	4·65	4·30	3·95	3·59	3·30	3·01	2·72
5½	140	5·77	5·33	4·89	4·52	4·15	3·77	3·47	3·16	2·85
5¾	146	6·05	5·59	5·11	4·73	4·34	3·95			
6	152	6·33	5·84	5·34						

TABLE 46 (*contd.*).—ALUMINIUM TUBE.

Weight in Lbs. per Lineal Foot.

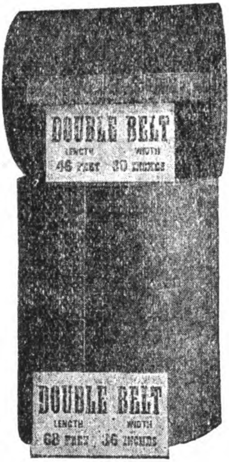
S. W. G. No.		10	11	12	13	14	15	16	17	18
Thickness Ins		0·128	0·116	0·104	0·092	0·080	0·072	0·064	0·056	0·048
„ Mm		3·25	2·94	2·64	2·34	2·03	1·83	1·62	1·42	1·22
Outside Dia. Ins.	Mm.									
$\frac{3}{16}$	4·76							0·029	0·027	0·025
$\frac{1}{4}$	6·35					0·050	0·047	0·044	0·041	0·036
$\frac{3}{8}$	9·52		0·111	0·104	0·096	0·087	0·080	0·073	0·066	0·058
$\frac{1}{2}$	12·7	0·176	0·165	0·152	0·139	0·124	0·113	0·102	0·092	0·080
$\frac{5}{8}$	15·9	0·235	0·219	0·200	0·181	0·161	0·146	0·132	0·116	0·101
$\frac{3}{4}$	19·0	0·294	0·270	0·246	0·223	0·199	0·180	0·162	0·143	0·124
$\frac{7}{8}$	22·2	0·353	0·325	0·297	0·267	0·235	0·216	0·192	0·170	0·146
1	25·4	0·412	0·379	0·345	0·309	0·272	0·247	0·221	0·197	0·168
$1\frac{1}{8}$	28·6	0·472	0·433	0·392	0·352	0·309	0·280	0·250	0·221	0·190
$1\frac{1}{4}$	31·8	0·531	0·486	0·441	0·393	0·346	0·314	0·282	0·246	0·212
$1\frac{3}{8}$	34·9	0·590	0·540	0·489	0·436	0·382	0·347	0·310	0·270	0·235
$1\frac{1}{2}$	38·1	0·649	0·594	0·537	0·479	0·420	0·380	0·340	0·300	0·260
$1\frac{5}{8}$	41·3	0·708	0·648	0·584	0·522	0·457	0·413	0·370	0·323	0·280
$1\frac{3}{4}$	44·5	0·767	0·701	0·632	0·564	0·494	0·446	0·398	0·350	0·300
$1\frac{7}{8}$	47·6	0·826	0·754	0·681	0·605	0·530	0·480	0·426	0·376	0·323
2	50·8	0·882	0·804	0·728	0·648	0·566	0·510	0·458	0·400	0·344
$2\frac{1}{8}$	53·9	0·945	0·880	0·776	0·690	0·605	0·541	0·494	0·423	0·376
$2\frac{1}{4}$	57·1	1·00	0·913	0·825	0·735	0·642	0·575	0·511	0·453	0·388
$2\frac{3}{8}$	60·3	1·06	0·968	0·872	0·776	0·681	0·598	0·552	0·470	0·411
$2\frac{1}{2}$	63·5	1·12	1·02	0·921	0·820	0·717	0·625	0·576	0·506	0·435
$2\frac{5}{8}$	66·6	1·18	1·08	0·970	0·862	0·752	0·658	0·611	0·517	0·458
$2\frac{3}{4}$	69·8	1·24	1·13	1·01	0·900	0·789	0·705	0·635	0·550	0·475
$2\frac{7}{8}$	73·0	1·30	1·18	1·07	0·946	0·823	0·728	0·670	0·576	0·505
3	76·2	1·35	1·23	1·11	0·989	0·875	0·770	0·690		
$3\frac{1}{4}$	82·6	1·47	1·34	1·20	1·06	0·935	0·835	0·750		
$3\frac{1}{2}$	88·9	1·59	1·44	1·30	1·15	1·00	0·900	0·800		
$3\frac{3}{4}$	95·2	1·71	1·55	1·40	1·25	1·19				
4	102	1·83	1·67	1·50	1·32					
$4\frac{1}{4}$	108	1·95	1·77	1·59	1·41					
$4\frac{1}{2}$	114	2·07	1·88	1·69						
$4\frac{3}{4}$	121	2·19	1·99	1·79						
5	127	2·31	2·10	1·88						
$5\frac{1}{4}$	133	2·43	2·20	1·98						
$5\frac{1}{2}$	140									
$5\frac{3}{4}$	146									
6	152									

TABLE 47.—WEIGHT OF DURALUMIN TUBES IN LBS. *per foot run.*

Gauge	Weight of Duralumin Tubes in lbs. per foot run													
	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2
10	·1228	·1859	·2495	·3128	·376	·440	·5032	·554	·617	·6786	·741	·801	·863	·926
11	·1159	·173	·230	·2861	·343	·400	·4565	·514	·5705	·628	·685	·741	·798	·8555
12	·1099	·1614	·213	·2648	·316	·368	·420	·4713	·5234	·575	·626	·678	·729	·782
13	·1009	·1456	·1909	·236	·2809	·3258	·3708	·416	·460	·506	·551	·5956	·641	·6855
14	·092	·1312	·1703	·210	·2492	·2885	·328	·367	·406	·446	·485	·5245	·564	·603
15	·08275	·1168	·151	·185	·219	·253	·2875	·3216	·3557	·3895	·424	·458	·492	·526
16	·07625	·1073	·1378	·1686	·1992	·230	·2609	·2918	·322	·3535	·3845	·415	·4455	·4765
17	·0698	·0975	·1245	·152	·1791	·2069	·2342	·262	·2894	·3168	·3441	·372	·3992	·427
18	·0605	·0838	·107	·1302	·1532	·1765	·200	·223	·2461	·2673	·2892	·312	·335	·358
19	·053	·073	·0927	·1128	·1323	·1524	·1725	·1922	·204	·2233	·2422	·2578	·2806	·2997
20	·0451	·0615	·0784	·095	·1112	·1282	·1445	·1611	·1843	·2015	·2305	·236	·253	·2702
21	·0415	·057	·072	·087	·102	·1173	·1323	·148	·1643	·176	·1948	·210	·2256	·2405
22	·0369	·0501	·0634	·0766	·09	·1031	·1163	·1296	·1441	·1574	·1708	·1843	·1976	·2104

SECTION IV

ENGINEERING DATA FOR TRANSMISSION OF POWER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GEARS AND GEARING.

Where rotary motion is required to be transmitted with accuracy, it is necessary to employ toothed gearing, which can be mainly divided into two classes, for use with parallel shafts and for use with shafts at an angle. The first class is comprised of spur, single helical and double helical gearing. The second class is comprised of bevel, worm, and spiral gearing.

Spur gears can be made either in one piece, or of rawhide, fibre or paper, the object in using the latter three varieties, which are made up in laminations, being to reduce noise to a minimum, but it is necessary that these have steel or gunmetal flanges with rivets or bolts through the whole.

Mortice wheels have wood teeth (preferably hornbeam) inserted and wedged into cast-iron or cast steel centres, the object of this design being that, while quiet in running, should any individual tooth break it can easily be replaced.

In modern practice, however, rawhide, fibre, paper, and mortice gears are rapidly being discarded in favour of double helical gears, which are equally silent in motion, have much longer life, and are now cheaper in manufacture owing to the cutting of this type of gear having been made the subject of much scientific investigation during the last few years.

In the past, many gears were made with machine-moulded teeth ; but the requirements of the geared turbine and other high-speed machinery, where the pinion is frequently run at 4,000 r.p.m., have made it necessary to employ accurate generating machinery in order to obtain gears to run at the required high speeds with safety and a high efficiency, it now being possible to produce commercially, gears to work under load with an efficiency of as much as 97 per cent.

For exceptionally high-speed gearing, it is necessary to employ special methods of mounting and lubrication, which are dealt with in the following pages.

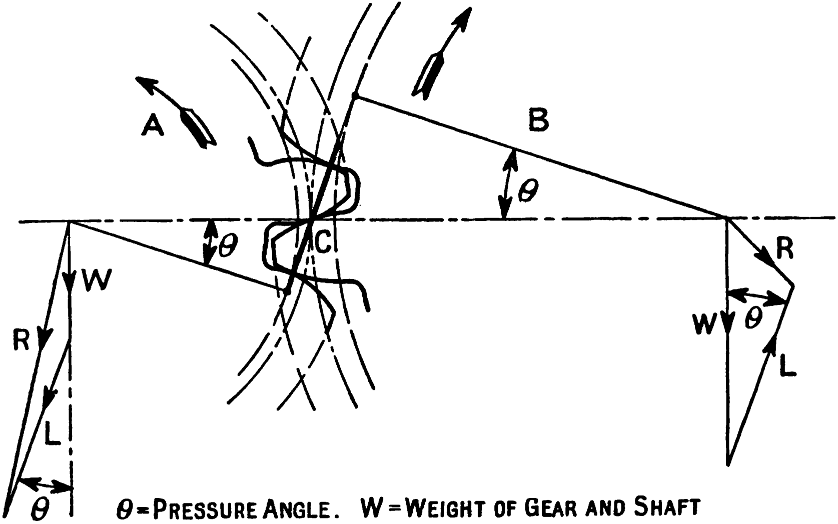
Recent investigations have proved that for turbine gearing it is advisable to use much smaller pitches than have been considered safe in the past, the present standards varying from ⅜ in. to a maximum of 1¼ ins. circular pitch.

For automobile rear-axle drives, the chain drives have been almost entirely superseded by gear drives, in the first cases by bevel gears and, later, by spiral bevel and worm gearing, this latter now being adopted for all types, including heavy lorry rear axles, on account of the high efficiency now obtainable.

PRESSURE ANGLE.

Pressure Angle is the angle at which the load acts on the flanks of the teeth and, by setting out the tooth profiles and drawing this line of pressure, the number of teeth in simultaneous contact can be determined as shown in Fig. 1.

In the case of spurs and similar gears, the pressure angle is



θ = PRESSURE ANGLE. W = WEIGHT OF GEAR AND SHAFT
L = LOAD ON TEETH. C = LINE OF CONTACT R = RESULTANT LOAD ON BEARINGS

FIG. 1.—FORCES IN GEARING.

the angle between a line tangent to the flank of a tooth at the pitch line and a radial line drawn from the intersection of the tooth profile and the pitch line.

In a worm, the angle of pressure can be measured on the linear or axial section from the outside of the worm, as shown in Fig. 2.

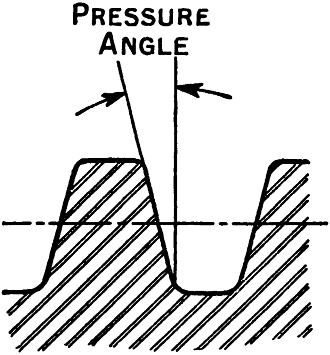


FIG. 2.—PRESSURE ANGLE.

The load on the teeth can be determined from the formula $\frac{\text{H.P.} \times 63{,}000}{\text{Revs. per min.} \times \text{Cos press. angle} \times \text{Radius}}$ and acts in the directions shown in Fig. 1, where pinion "A" is the driver and wheel "B" the driven member. Where wheel "B" becomes the driver, these conditions are reversed.

The standard angles of pressure in general use are—

General spur gearing	14½°
Spurs for automobile and similar high-speed mechanism	20°
Turbine reduction gearing	20°
General worm gearing	15°
Worm gearing for exceptional high speed or efficiency	22½°–30°
Bevel gearing	14½°
Automobile bevel gearing	20°

Where gears are to be cut for use as patterns and the pinion is of 20 teeth or less, it is necessary to adopt a pressure angle of 20°, as, if these are made 14½° and the flanks correspondingly undercut, there is a tendency for the teeth to pull away the sand when withdrawing the pattern.

LUBRICATION.

Where rawhide or mortice gearing is installed, it should be remembered that oil must not be used as a lubricant, as it softens the material and considerably shortens its life. For this class of gear it is advisable to use graphite or some of the various special preparations now on the market.

For other types of gearing, the speed must be considered in determining the method to be used. For tangential speeds up to 1,000 ft. per min., the gears may be allowed to dip in an oil bath; but where the tangential speed of the periphery of the gears exceeds 1,000 ft. per min., the oil must be forced into the advance point of engagement of the gears under a pressure of from 5 to 8 lbs. per square inch. If the gears were allowed to dip, the churning effect of the wheels would break up the oil.

In order to obtain the necessary oil spray, special types of oil nozzles must be used, the object of these being to obtain an oil discharge of a "fish tail" shape and so distribute the oil over the full area of engagement, a suitable number of nozzles being fitted to make the total width of the sprays equal to the face width of the gears.

The design of these nozzles has been the subject of considerable investigation, the latest types being illustrated below.

The diameter of the hole A and the width of the slot B is varied from $\frac{1}{16}$ in. to $\frac{1}{8}$ in. to suit the viscosity of the oil adopted.

For the lubrication of worm gearing, it is advisable to adopt an oil bath fitted with an oil level gauge to ensure the oil not reaching above the axis of the worm, it not being advisable to allow the teeth of the wheel to dip in the oil.

The teeth of all gears should be cut 0·005 ins. to 0·010 in. thinner on the pitch line than the theoretical standard of half the pitch, in order to allow of a suitable oil film, when the load is transmitted by the film and not by the contact of the metals: this reduces the coefficient of friction and, therefore, increases the efficiency of the gears and also lengthens their life if an oil is obtained which will not break up under the pressure to be applied.

There are now numerous oils on the market specially manufactured to endure under these conditions.

MOUNTING.

Spur-gearing. When cutting spur gears, it is good practice to scribe the pitch lines on the sides of the blanks, as this enables the gears to be mounted to give correct running clearance between the teeth. Bearings should be as close to the gears as possible, and each gear should have bearings at either side, the life of gears being considerably reduced by being overhung from one bearing. It is necessary to locate one bearing on each shaft in order to keep the gears correctly in mesh.

Double helical-gearing should be mounted in a similar manner to spur gears, with the exception that only one bearing be located in each pair of gears, preferably on the wheel shaft; the pinion should be driven by means of a flexible coupling allowing a certain amount of lateral "float": this enables the gears to find their own centre and thus prevents one-half of the gear taking all the load and also preventing any external thrust coming on the gears with a similar effect. Where coupled to a fan or similar external unit giving a longitudinal

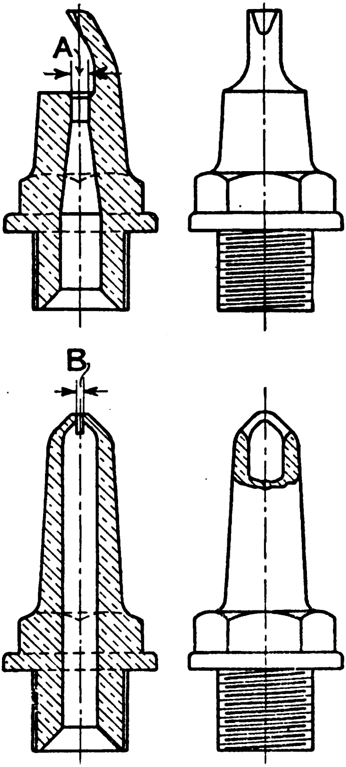


FIG. 3.—NOZZLES FOR GEAR LUBRICATION.

thrust, a thrust bearing should be installed on the shaft concerned, as otherwise the located bearing would heat up and give trouble.

In the case of turbine or other exceptionally high-speed drives, an oil-tight case should be supplied capable of confining the inflammable gases given off by the oil, and a suitable ventilator provided at the top of the case with baffles to prevent oil being discharged into the atmosphere in its inflammable state. In these cases, the flexible coupling also allows the shafts to expand under the heat without creating distortion.

Worm and Spiral-gearing. Thrust bearings must be used on all shafts. Where the direction of rotation is constant, a single thrust washer or flanged bearing may be used, according to the thrust to be taken; but where the gear is reversible, a double-thrust bearing must be used. The necessary position of the thrust washer can be obtained from Fig. 4.

Where worm gears are used for winding or haulage gears, the angle of the worm thread should not exceed 13°, as above this angle it is possible for the gear to run back (*i.e.* for the wheel to drive the worm).

Worm gears should run with the worm dipping in an oil bath to a depth equal to the depth of the worm thread, and an oil-level gauge provided on the outside of the case to ensure the maintaining of this level.

Bevel-gearing should always be provided with a single-thrust bearing immediately behind each gear (*i.e.* adjacent to the large end of the teeth). Bevel gears should be mounted with a bearing on each side of each gear, but where unavoidable, it is allowable in this type of gear to run the pinion overhung with one bearing of ample size on the same side as the thrust washer.

PROCESSES OF MANUFACTURE.

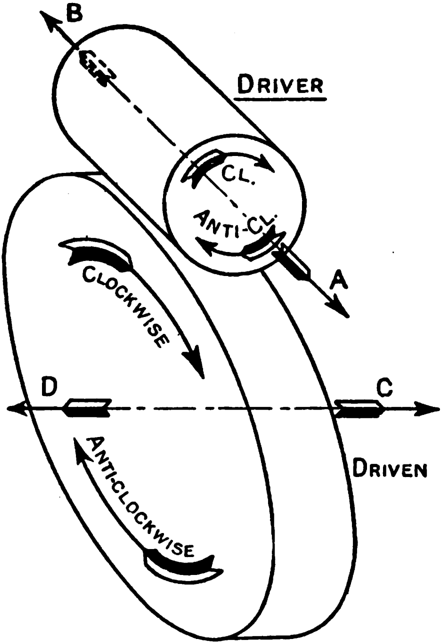
Spur gears are usually milled, small quantities being conveniently cut on an ordinary milling machine; but where large numbers are required, it is better to use a single purpose machine, where greater facilities are given for dividing, etc.

Milling cutters are made each to cut within a certain range—the most general being:

Cutter	No. 1 ..	135 t.–rack	Cutter	No. 5 ..	21 t.–25 t.
,,	No. 2 ..	55 t.–134 t.	,,	No. 6 ..	17 t.–20 t.
,,	No. 3 ..	35 t.– 54 t.	,,	No. 7 ..	14 t.–16 t.
,,	No. 4 ..	26 t.– 34 t.	,,	No. 8 ..	12 t.–13 t.

For mass production of small gears, a "Fellows" gear-shaper is employed to considerable advantage. The cutter is of pinion form, with teeth ground to correct profiles. This cutter acts with a slotting and generating action.

Large pitch spurs are frequently planed on a machine specially adapted for this work, the profile being planed by a single point tool carried by a cutter slide acting by means of a roller on to a former of the required profile. This method is

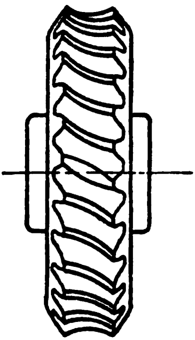


HAND OF THREAD	DRIVER	DRIVEN	THRUST DIRECTIONS
RIGHT	CLOCKWISE	CLOCKWISE	B & D
RIGHT	ANTI-CLOCKWISE	ANTI-CLOCKWISE	A & C
LEFT	CLOCKWISE	ANTI-CLOCKWISE	A & D
LEFT	ANTI-CLOCKWISE	CLOCKWISE	B & C

FIG. 4.—DIRECTION OF ROTATION OF WORM AND SPIRAL GEARS.

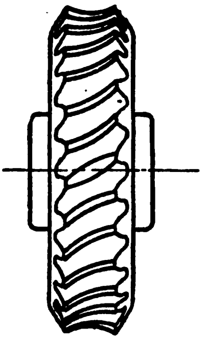
particularly suitable for cutting special shapes in small quantities, the former being cheaper than a special milling cutter.

Worm Gears. The wheel is cut by a hob, the thread of which is chased to the shape of worm thread adopted, and then gashed and relieved to the correct cutting angle. The hob is tapered to give a gradually increasing cutting depth while feeding through the blank. When a hob is not available, a fly-cutter is used with a cutting face of the same profile as the normal section of the worm thread, mounted in a body slightly less in diameter than the root of the worm and with the cutter face normal to the worm thread angle. The worm can be either milled or chased in the lathe.



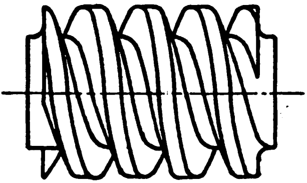
RIGHT HAND WHEEL

FIG. 5.



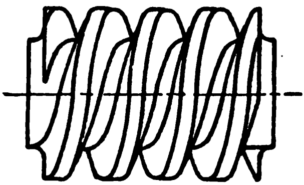
LEFT HAND WHEEL

FIG. 6.



RIGHT HAND WORM

FIG. 7.



LEFT HAND WORM

FIG. 8.

Spiral gears can be milled with a spur cutter made to cut a number of teeth equal to

$$\frac{\text{No. of teeth in spiral}}{\text{Cos}^3 \text{ spiral angle}}$$

or they may be hobbed with a parallel hob of the same normal pitch as the gear to be cut.

Double helical gears can be end-milled with a formed cutter, or planed with a cutter either of rack formation or similar to a "Fellows" cutter, these last two generating the tooth profile. They also can be hobbed as spiral gears if a centre gap is left equal to 1¼ × Circular pitch.

Bevel gears can be planed as described for spurs, but the former profile must be enlarged in the proportion of

$$\frac{\text{Distance of machine}}{\text{Distance of bevel}}$$

Small bevels are generally cut by a planing and generating process with a triangular cutter, with an included angle equal to twice the pressure angle.

Bevels are not satisfactory when milled, except where the pitch angle is very small, and even then the gears run noisily at high speeds.

MATERIALS.

A comparison of the strengths of materials used in the manufacture of gearing can be made from the following table, in which cast iron is taken as a basis—

Cast iron	1·0
High carbon steel	4·0–5·0
Forged steel	3·0
Cast steel	2·2
Phosphor bronze	1·6
Rawhide	0·8
Hornbeam (for mortice wheels)	0·8
Compressed paper	0·7

In the case of gears working under heavy load, it is not advisable to make two gears of like material.

Suitable combinations in general use are as follows—

Cast iron	with	forged steel ;
,, ,,	,,	rawhide, paper, or hornbeam ;
,, ,,	,,	cast steel ;
Forged steel	,,	high carbon steel ;
,, ,,	,,	rawhide, paper, or hornbeam ;
Phosphor bronze	,,	forged steel (preferably case-hardened)
,, ,,	,,	high carbon steel.

For turbine reduction gear pinions where shafts are kept as small as possible to keep down the rubbing speed on the periphery, it is usual to use nickel or nickel-chrome steel, and subject this to an oil-toughening process after first rough machining the pinion to within about half an inch of finished dimensions.

For worm-gearing, the highest efficiency is obtained by mating a phosphor bronze worm wheel with a low-carbon, forged steel worm, which has been case-hardened, and the threads afterwards ground to remove any distortion and also give a glassy surface. This gives a very low coefficient of friction and reduces to a minimum the amount of waste energy given off in heat.

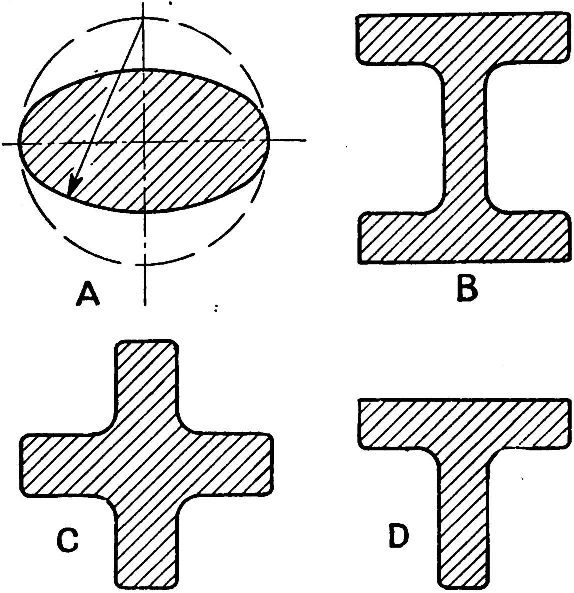


Fig. 9.—Wheel Arm Sections.

Wheel Arms.

The number of arms required for any type of gear can be ascertained from the following table—

Up to 1′–0″ diam.	Plate wheel
1′–0″ diam. to 2′–0″ diam.	4 arms
2′–0″ ,, 3′–0″ ,, ...	5 ,,
3′–0″ ,, 8′–0″ ,, ...	6 ,,
8′–0″ ,, 16′–0″ ,, ...	8 ,,

Oval arms, as shown in section at A (Fig. 9), are only suitable for light duties, such as change gears, etc., where, generally speaking, no considerable load is applied.

"T" section arms, as shown at D (Fig. 9), are usually confined in their application to bevel wheels.

"H" section arms, as shown at B (Fig. 9), are suitable for spur, helical, or worm wheels when they are required to sustain considerable loads.

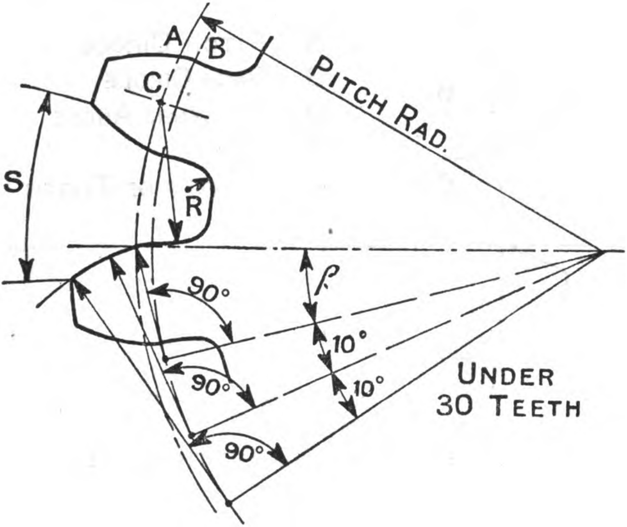


FIG. 10.—THE SETTING OUT OF CYCLOIDAL GEAR TEETH.

"+" section arms, as shown at C (Fig. 9), are sometimes used for similar purposes as the "H" section; but the use of the latter is preferable owing to its greater stiffness under working conditions.

For turbine reduction gear wheels, the design is usually of the plate-wheel type, as this lends itself to being machined all over and, therefore, is much easier balanced; and it is also possible to more easily detect flaws in the material, which would be fatal to gears running at exceptionally high speeds as in these cases.

It is first necessary to determine the diameter of the rolling circle required. Where there is a small pinion in the train of gears under consideration, it is desirable to use a rolling circle equal in diameter to half the pitch diameter of the pinion, as this gives this pinion radial flanks and so prevents excessive undercut.

Where a small pinion is not employed, it is usual to make the rolling circle equal in diameter to half the pitch diameter of an assumed 15-teeth pinion, and this is frequently adopted as a standard.

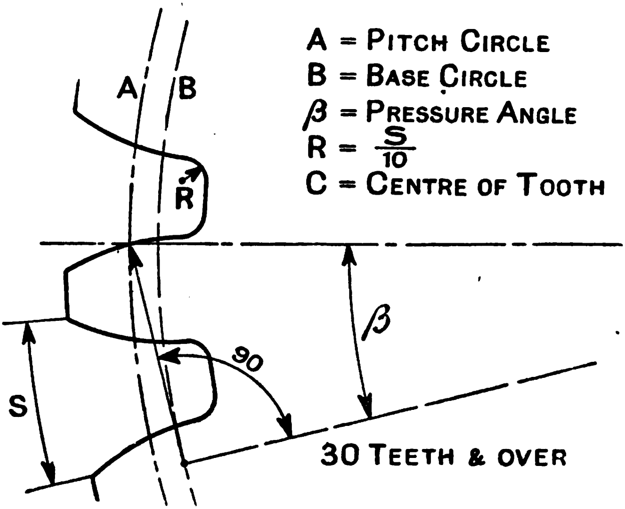


Fig. 11.—The Setting Out of Cycloidal Gear Teeth.

Cycloidal Teeth.

To draw the epicycloid which forms the profile above the pitch line, take periodic positions of the rolling circle with its centre at *a*, *b*, *c*, *d*, *e*, *f*, and draw a portion of each circle. Then mark of *g* A1 equal to *g* A, *h* A2 equal to *h* A, *j* A3 equal to *j* A, etc., until sufficient positions to form the upper portion of the profile are obtained.

In a similar manner, draw the hypocycloid forming the portion of the profile below the pitch line.

The line of contact between this gear and its mate will always lie along the line BAC, the length of the portion AC varying according to the diameter of the mating gear.

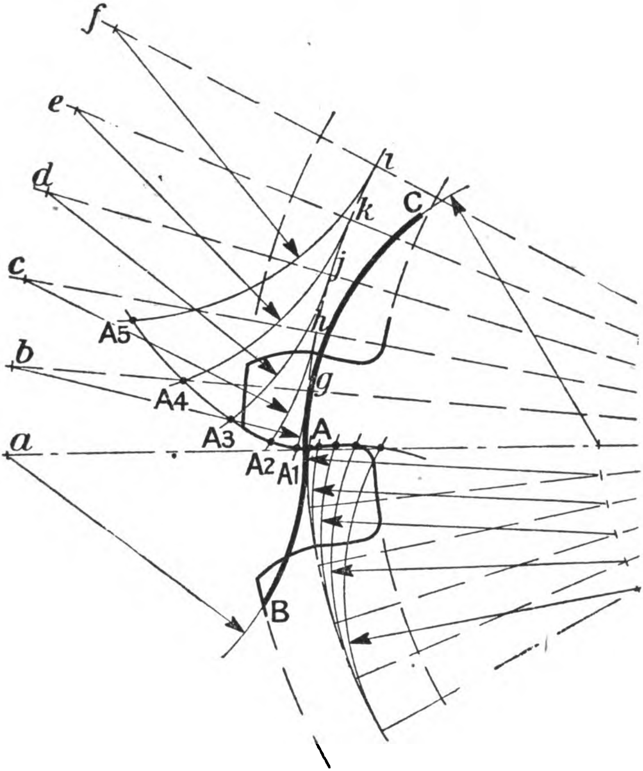


FIG. 12.—THE SETTING OUT OF CYCLOIDAL GEAR TEETH, SHOWING LINE OF CONTACT.

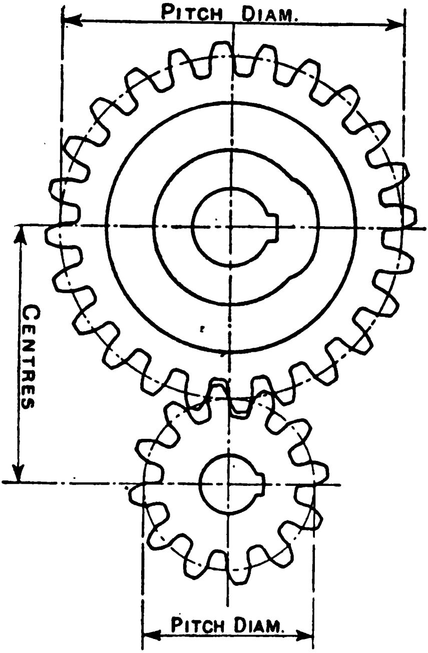


FIG. 13.—SPUR GEARS (Centres).

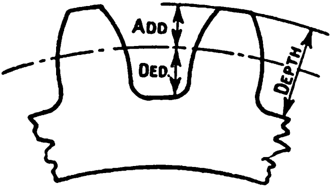


FIG. 14.—SPUR GEARS (Depth of Teeth).

SPUR GEARS.

Particulars required.	Particulars given.	Formulae.
No. of teeth	Pitch diam. and diametral P.	Pitch diam. × Diametral P.
No. of teeth	Outside diam. & diam. P.	Outside diam. × Diam. P − 2
No. of teeth	Pitch diam. & circular P.	$\frac{\text{Pitch diam.} \times \pi}{\text{Circular pitch}}$
No. of teeth	Outside diam. & circular P.	$\frac{\text{Outside diam.} \times \pi}{\text{Circular pitch}} - 2$
Diametral pitch	Pitch diam. & no. of teeth	$\frac{\text{No. of teeth}}{\text{Pitch diam.}}$
Diametral pitch	Outside diam. & no. of teeth	$\frac{\text{No. of teeth} + 2}{\text{Outside diam.}}$
Circular pitch	Pitch diam. & no. of teeth	$\frac{\text{Pitch diam.} \times \pi}{\text{No. of teeth}}$
Circular pitch	Outside diam. & no. of teeth	$\frac{\text{Outside diam.} \times \pi}{\text{No. of teeth} + 2}$
Outside diam.	No. of teeth & diam. P.	$\frac{\text{No. of teeth} + 2}{\text{Diam. pitch}}$
Outside diam.	No. of teeth & circular P.	$\frac{\text{Circular P.} \times (\text{No. of teeth} + 2)}{\pi}$
Pitch diam.	No. of teeth & diam. P.	$\frac{\text{No. of teeth}}{\text{Diam. pitch}}$
Pitch diam.	No. of teeth & circular P.	$\frac{\text{Circular P.} \times \text{No. of teeth}}{\pi}$
Addendum	Diametral P.	$\frac{1}{\text{Diam. pitch}}$
Addendum	Circular P.	Circular P. × ·3183
Dedendum	Diametral P.	$\frac{1{\cdot}157}{\text{Diam. pitch}}$
Dedendum	Circular P.	Circular P. × ·3683
Full depth of tooth	Diametral P.	$\frac{2{\cdot}157}{\text{Diam. pitch}}$
Full depth of tooth	Circular P.	Circular P. × ·6866
Centre distance	Diam. P. & nos. of teeth in both gears	$\frac{\text{Sum of nos. of teeth}}{\text{Diam. pitch} \times 2}$
Centre distance	Circular P. & nos. of teeth in both gears	$\frac{\text{Sum of nos. of teeth} \times \text{C.P.}}{\pi \times 2}$

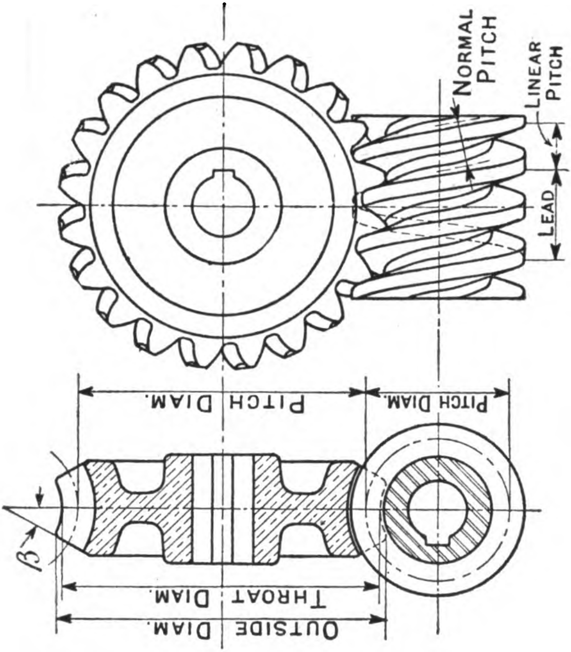


FIG. 15.—WORM GEARS.

WORM GEARS.

Particulars required.	Particulars given.	Formulae.
Circular P. of wheel or linear P. of worm	Pitch diam. of wheel & no. of teeth in wheel	$\frac{\text{Pitch diam.} \times \pi}{\text{No. of teeth in wheel}}$
Pitch diam. of wheel	No. of teeth & circular P.	$\frac{\text{No. of teeth} \times \text{Circular P.}}{\pi}$
Throat diam. of wheel	Pitch diam. & circular P.	Pitch diam. + ·6366 × CircularP
Pitch diam. of worm	Pitch diam. of wheel & centre distance	Centres × 2 − P. diam. wheel
Lead of worm	No. of threads in worm and linear pitch	No. of threads × Linear P.
Lead angle of worm	Worm P. diam. and lead	$\frac{\text{Lead}}{\text{Pitch diam.} \times \pi} = \text{Tan}$
Addendum	Circular pitch	Circular pitch × ·3183
Dedendum	Circular pitch	Circular pitch × ·3683
Depth of tooth	Circular pitch	Circular pitch × ·6866
Worm outside diam.	Worm P. diam. & addendum	P. diam. + 2 × Addendum
Worm root diam.	Worm P. diam. & dedendum	P. diam. − 2 × Dedendum
Centre distance	P. diams. of worm & wheel	$\frac{\text{Sum of pitch diams.}}{2}$
Maximum length: Worm	Throat diam. wheel and addendum	$\sqrt{\text{Throat diam.}^2 - (\text{Throat diam.} - 4 \text{ add.})^2}$
Radius of wheel throat	Pitch diam. of worm and addendum	$\frac{\text{P. diam. worm} - 2 \text{ ad.}}{2}$

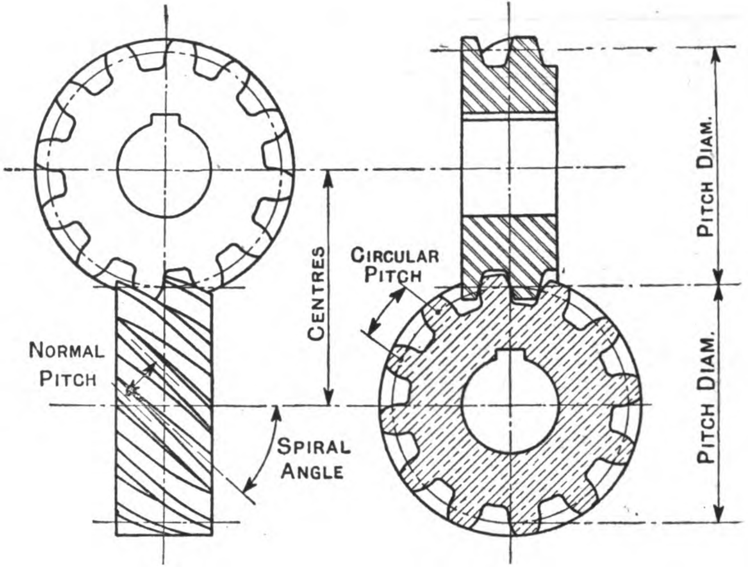


FIG. 16.—SPIRAL GEARS.

SPIRAL GEARS.

Particulars required.	Particulars given.	Formulae.
Circular P.	Pitch diam. & no. of teeth	$\frac{\text{Pitch diam.} \times \pi}{\text{No. of teeth}}$
Circular P.	Outside diam. & spiral angle	$\frac{\text{Outside diam.} \times \pi}{\text{No. of T.} + 2 \cos \text{spiral angle}}$
P. diam.	Circular P. & no. of teeth	$\frac{\text{Circular P.} \times \text{no. of teeth}}{\pi}$
Spiral angle wheel	Circular P. of wheel & pinion	$\frac{\text{Circular P. wheel}}{\text{Circular P. pinion}} = \text{Tan}$
Spiral angle pinion	Circular P. of wheel & pinion	$\frac{\text{Circular P. pinion}}{\text{Circular P. wheel}} = \text{Tan}$
Normal P.	Circular pitch & spiral angle	Circular P. × Cos. spiral angle
Lead of spiral	Pitch diam. & spiral angle	$\frac{\text{Pitch diam.} \times \pi}{\text{Tan spiral angle}}$
No. of teeth for which to select cutter	No. of teeth in gear to be cut & spiral angle	$\frac{\text{No. of teeth}}{\text{Cos}^3 \text{ spiral angle}}$
Addendum	Normal pitch	Normal pitch ×·3183
Dedendum	Normal pitch	Normal pitch ×·3683
Depth of tooth	Normal pitch	Normal pitch ×·6866
Centre distance	P. diams. of wheel & pinion	$\frac{\text{Sum of pitch diams.}}{2}$
Outside diameter	Pitch diam. & addendum	Pitch diam. + 2 add.
Normal tooth thickness	Circular P. & spiral angle	$\frac{\text{Circular P.} \times \text{Cos. spiral angle}}{2}$

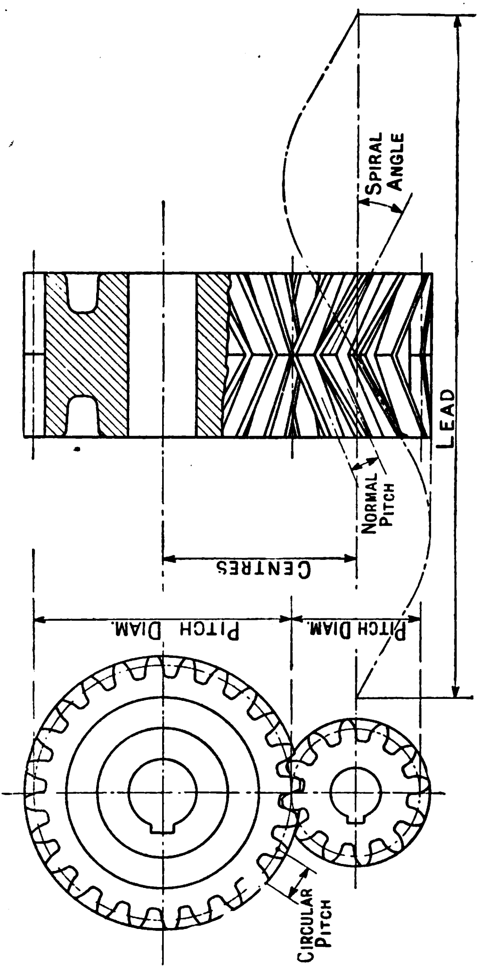


Fig. 17.—Helical Gears.

HELICAL GEARS.

Particulars required.	Particulars given.	Formulae.
P. diam.	No. of teeth & circular pitch	$\frac{\text{No. of teeth} \times \text{pitch}}{\pi}$
P. diam.	No. of teeth & diametral P.	$\frac{\text{No. of teeth}}{\text{Diametral pitch}}$
Outside diameter	Pitch diam. & addendum	Pitch diam. + 2 add.
Normal P.	Circular pitch spiral angle	Circular P. × Cos. spiral angle
Addendum	Normal pitch	Normal pitch ×·3183
Dedendum	Normal pitch	Normal pitch ×·3683
Depth of tooth	Normal pitch	Normal pitch ×·6866
No. of teeth for which to select cutter	No. of teeth in gear to be cut & spiral angle	$\frac{\text{No. of teeth}}{\text{Cos}^3 \text{ spiral angle}}$
Lead of spiral	Pitch diam. & spiral angle	$\frac{\text{Pitch diam.} \times \pi}{\text{Tan.}: \text{spiral angle}}$
Angle of spiral	—	22½°, 30°, or 45°, according to process of manufacture
Centre distance	Pitch diams. of wheel & pinion	$\frac{\text{Sum of pitch diams.}}{2}$
Normal tooth thickness	Circular pitch & spiral angle	$\frac{\text{Circular P.} \times \text{Cos spiral angle}}{2}$

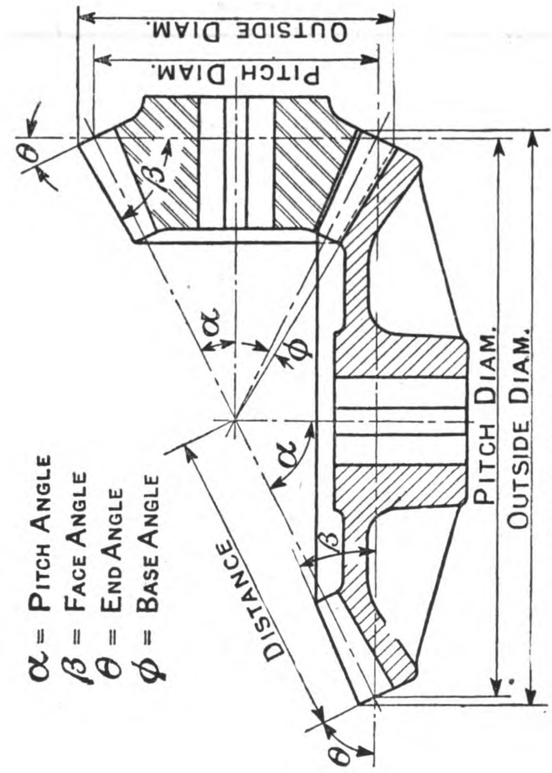


Fig. 18.—Bevel Gears.

BEVEL GEARS.

Particulars required.	Particulars given.	Formulae.
Back angle and pitch angle : Wheel	Nos. of teeth in wheel & pinion	$\frac{\text{No. of teeth : Wheel}}{\text{No. of teeth : Pinion}} = \text{Tan}$
Back angle & P. angle pinion	Nos. of teeth in wheel & pinion	$\frac{\text{No. of teeth : Pinion}}{\text{No. of teeth : Wheel}} = \text{Tan}$
Pitch diameter	No. of teeth & circular P.	As for spurs
Pitch diameter	No. of teeth & diam. P.	
Distance	Pitch diams. of wheel & pinion	$\sqrt{\text{Wheel P.diam.}^2 + \text{Pinion P. diam.}^2}$
Distance	Pitch diam. & pitch angle	$\frac{\text{Pitch diam.}}{\text{2 sin pitch angle}}$
Top angle	Addendum and distance	$\frac{\text{Addendum}}{\text{Distance}} = \text{Tan}$
Base angle	Dedendum & distance	$\frac{\text{Dedendum}}{\text{Distance}} = \text{Tan}$
Face angle	Pitch angle & top angle	90°– (P. angle + Top angle)
Outside diameter	Pitch diam. : Pitch angle & addendum	P. diam. + 2 cos. spl. angle × Add.
Addendum Dedendum Depth of tooth	Circular pitch or diametral pitch	As for spurs
Circular P.	Outside diam. : No. of teeth & pitch angle	$\frac{\text{Outside diam.} \times \pi}{\text{No. of T.} + \text{2 cos. : P. angle}}$

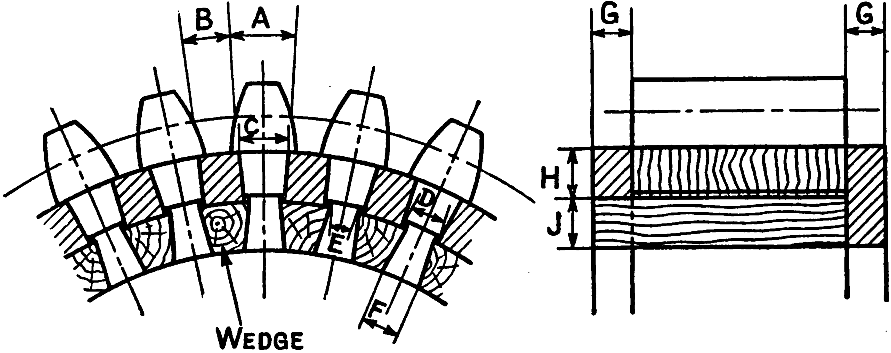


FIG. 19.—RAWHIDE SPUR GEARS.

$A = \cdot 56 \times$ Pitch.
$B = \cdot 44 \times$ Pitch.
$C = \cdot 50 \times$ Pitch.

$D = \cdot 40 \times$ Pitch.
$E = \cdot 30 \times$ Pitch.
$F = \cdot 40 \times$ Pitch.

$G = \cdot 40 \times$ Pitch.
$H = \cdot 50 \times$ Pitch.
$J = \cdot 50 \times$ Pitch.

RAWHIDE SPUR GEARS.

Particulars required.	Particulars given.	Formulae.
Tooth particulars	—	As for ordinary spur gears
Face width	Face width of mating gear	Face of mate + $\frac{1}{4}''$
Thickness of flanges	Circular pitch	$\frac{\text{Circular pitch}}{3}$
Diameter of rivets	Circular pitch & no. of teeth	$\cdot 01 \times \text{Pitch} \times \text{No. of teeth}$
No. of rivets	Rivet circle diam. & diam. of rivets	$\frac{\text{Rivet circle diam.}}{2 \times \text{Rivet diam.}}$
Minimum diam. of bush	Bore diam.	$2 \times \text{Bore diam.}$
Rivet circle diam.	Root and Bush diams.	$\frac{\text{Root diam.} + \text{Bush diam.}}{2}$

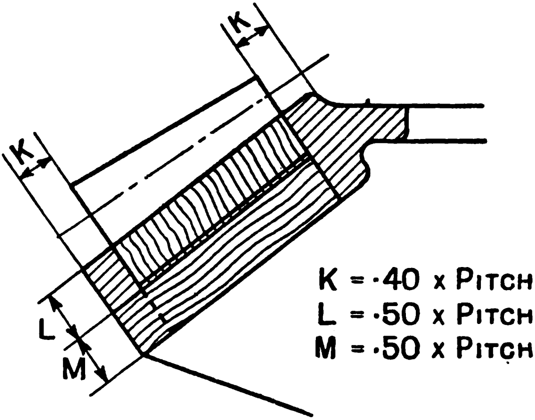


Fig. 20.—Rawhide Bevel Gears.

Particulars required.	Particulars given.	Formulae.
Tooth particulars	—	As for ordinary bevels
Angle for drilling for rivets	Pitch angle & base angle	$\frac{\text{Pitch angle} + \text{Base angle}}{2}$
Thickness of front flange	Circular pitch	Circular pitch $\times \cdot 25$
Thickness of back flange	Circular pitch	Circular pitch $\times \cdot 4$
Rivet circle diam. at front end	Root diam. at front end & bush diam.	$\frac{\text{Root diam.} + \text{Bush diam.}}{2}$
No. & diam. of rivets	—	As for spurs working on pitch at back end

TABLE 1.—CONVERSION OF PITCHES.

Circular Pitch in Ins.	Diam-etral Pitch.	Metric Module.	Circular Pitch in Ins.	Diam-etral Pitch.	Metric Module.
0·1237	25·4000	1·0000	0·8040	3·9076	6·5000
1/8	25·1327	1·0106	13/16	3·8666	6·5688
0·1309	24·0000	1·0583	0·8658	3·6286	7·0000
0·1428	22·0000	1·1545	7/8	3·5904	7·0741
0·1571	20·0000	1·2700	0·8976	3·5000	7·2572
0·1653	19·0000	1·3368	15/16	3·3510	7·5794
0·1745	18·0000	1·4111	0·9895	3·1749	8·0000
0·1848	17·0000	1·4941	1	3·1416	8·0847
0·1855	16·9330	1·5000	1·0472	3·0000	8·4665
3/16	16·7552	1·5159	1·1132	2·8222	9·0000
0·1963	16·0000	1·5875	1 1/8	2·7925	9·0953
0·2094	15·0000	1·6933	1·1424	2·7500	9·2360
0·2244	14·0000	1·8143	1·2369	2·5400	10·0000
0·2417	13·0000	1·9538	1 1/4	2·5133	10·1058
0·2474	12·7000	2·0000	1·2566	2·5000	10·1598
1/4	12·5664	2·0212	1·3606	2·3090	11·0000
0·2618	12·0000	2·1166	1 3/8	2·2848	11·1164
0·2856	11·0000	2·3090	1·3963	2·2500	11·2887
0·3092	10·1598	2·5000	1·4842	2·1166	12·0000
5/16	10·0531	2·5265	1 1/2	2·0944	12·1270
0·3142	10·0000	2·5400	1·5708	2·0000	12·7000
0·3491	9·0000	2·8222	1·6079	1·9538	13·0000
0·3711	8·4665	3·0000	1 5/8	1·9333	13·1376
3/8	8·3776	3·0318	1·7316	1·8143	14·0000
0·3927	8·0000	3·1749	1 3/4	1·7952	14·1482
0·4329	7·2572	3·5000	1·7952	1·7500	14·5144
7/16	7·1808	3·5371	1·8553	1·6933	15·0000
0·4488	7·0000	3·6286	1 7/8	1·6755	15·1588
0·4947	6·3499	4·0000	1·9790	1·5875	16·0000
1/2	6·2832	4·0423	2	1·5708	16·1693
0·5236	6·0000	4·2332	2·0944	1·5000	16·9330
0·5566	5·6444	4·5000	2·2263	1·4111	18·0000
9/16	5·5851	4·5476	2 1/4	1·3962	18·1905
0·6184	5·0799	5·0000	2·4737	1·2700	20·0000
5/8	5·0265	5·0529	2 1/2	1·2566	20·2116
0·6283	5·0000	5·0799	2·5133	1·2500	20·3196
0·6803	4·6180	5·5000	2 3/4	1·1424	22·2328
11/16	4·5696	5·5582	2·9684	1·0583	24·0000
0·7421	4·2332	6·0000	3	1·0472	24·2540
3/4	4·1888	6·0635	3·0922	1·0160	25·0000
0·7854	4·0000	6·3499	3·1416	1·0000	25·3995

STRENGTH OF WORM GEARS.

$$\text{Horse-power} = \frac{C \times \sqrt{D} \times d \times \text{Tan}\,\beta \times S}{33{,}000}$$

C = Constant from Table 2 ;
D = Pitch diameter of wheel ;
d = Pitch diameter of worm ;
S = Tangential speed of wheel pitch line ;
β = Half angle of face. (*See* Fig. 15.)
End thrust of worm = Total pressure on wheel teeth

$$\text{Total pressure on wheel teeth} = \frac{\text{H.P.} \times 63{,}000}{\text{Revs. per min.} \times \text{wheel radius}}$$

TABLE 2.—HORSE POWER CONSTANT C.

S Feet per min.	C.	S Feet per min.	C.
50– 100	440	1,001–1,100	95
101– 200	407	1,101–1,200	88
201– 300	368	1,201–1,400	63
301– 400	329	1,401–1,600	50
401– 500	288	1,601–1,800	42
501– 600	246	1,801–2,000	35
601– 700	207	2,001–2,200	29
701– 800	170	2,201–2,400	26
801– 900	141	2,401–2,600	22
901–1,000	113	2,601–2,800	17

The above table gives values based on a bronze worm wheel running with a hardened steel worm. For other materials, work proportionately as for spur gears.

$$\text{Efficiency} = \frac{\text{Tan : Worm angle } (1 - \mu \text{ tan : Worm angle})}{\text{Tan worm angle} + \mu}$$

μ = Coefficient of friction = from ·020–·050.

FORMULAE FOR HORSE-POWER OF SPUR GEARS.

$$\text{Horse-power} = \frac{\text{Safe load in lbs.} \times \text{Feet per min.} \times \text{Pitch} \times \text{Face}}{33{,}000}$$

$$\text{Pitch at mid-face} = \frac{\text{Pitch} \times \text{Distance}}{\text{Distance} - \frac{\text{Face}}{2}}$$

$$\text{Velocity} = \frac{\text{Mean Pitch diam.} \times \text{Revs. per min.} \times \pi}{12} \text{ Ft. per min.}$$

$$\text{Mean pitch diam.} = \frac{\text{Pitch diam. : at large end} \times \text{Distance}}{\text{Distance} - \frac{\text{Face}}{2}}$$

TABLE 3.— SAFE LOAD IN LBS. FOR CAST IRON.

For other materials, multiply by factor given on page 249.

No. of Teeth	Speed of Pitch Line in Feet per Minute :													
	50 to 100.	**101 to 150.**	**151 to 200.**	**201 to 250.**	**251 to 300.**	**301 to 400.**	**401 to 500.**	**501 to 600.**	**601 to 800.**	**801 to 1,000.**	**1,001 to 1,200.**	**1,201 to 1,500.**	**1,501 to 1,800.**	**1,801 to 2,400.**
12	320	300	280	268	256	242	228	214	186	157	128	118	107	90
13	335	314	293	280	268	254	239	224	195	164	134	123	112	94
14	345	323	302	289	276	261	246	231	200	169	139	127	115	97
15	359	337	314	300	287	272	256	240	208	176	143	132	120	101
16	368	346	322	308	294	278	262	246	214	181	147	136	123	104
17	383	359	335	321	307	290	271	256	222	188	154	141	128	109
18	397	372	347	332	317	300	283	266	230	195	159	146	133	116
20	431	404	377	360	336	326	307	288	250	211	173	159	144	122
25	464	435	406	388	372	350	330	310	270	228	186	171	155	131
30	489	450	427	402	392	363	342	326	279	236	196	177	163	138
40	517	485	454	433	415	391	368	346	301	254	207	191	173	147
50	527	495	469	441	430	398	376	358	306	259	215	194	179	152
60	547	513	478	458	437	414	390	365	318	268	219	202	182	155
75	556	521	487	466	445	420	396	372	324	273	223	205	186	158
100	566	531	495	475	453	428	404	378	329	278	227	209	189	160
150	575	539	504	481	461	435	410	384	334	282	231	212	192	163
300	585	548	512	490	468	442	417	391	340	287	234	216	195	166
Rack	595	558	520	498	476	450	424	397	346	292	236	220	198	169

For double helical gears, add 10 per cent. to 15 per cent. on value obtained for spurs, according to nature of load.

For bevel gears, calculate as for spurs, but work on mean pitch and velocity at mid-face.

CHAINS AND CHAIN WHEELS.

A roller chain consists of inner and outer link elements, the former composed of two plates, two bushes and two rollers, the latter of two plates and two shouldered rivets.

The plates form the connections and housings for the wearing components of the chain, and their form and dimensions are designed so that at the pierced portion of the plates their breaking load is in proportion to that of the rivets.

They are manufactured from steel of high tensile strength.

Rivets are manufactured from nickel steel rods, which are ground truly round and to size, and hardened.

Bushes are fitted into the plates of the inner link, their internal surface forming the bearing for the full length of the barrel of the rivet and their external surface the bearing for the rollers.

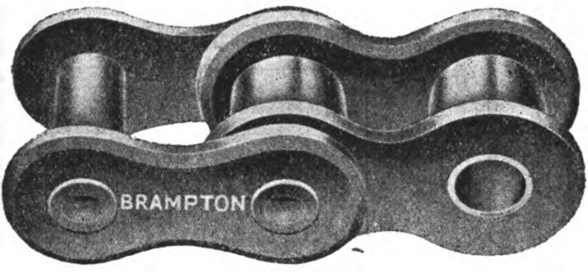


FIG. 21.—ROLLER CHAIN.

Bushes are machined from rods of mild steel, and are case-hardened all over, gauged for length and internal and external diameters.

The rollers are a component of the inner link, mounted on the bush on which they are free to revolve. Their movement reduces friction and wear between chain and wheels which would otherwise occur when the chain is moving on to or off the wheel teeth.

Rollers are machined from a special grade of mild steel and case-hardened all over.

INVERTED TOOTH-TYPE CHAINS.

Inverted tooth-type chains are made up of a series of flat plates with angular projections, which, ground to angles of 60°, and in some pitches of chain 75°, form the driving faces of the chain. These plates are fitted with hardened steel bushes in the pierced holes, and assembled on hardened steel rivets.

The plates are designed so that at their pierced portion they have a breaking load relative to that of the rivet, while providing the greatest possible diameter for the bush which forms the bearing surface for the rivet. They are manufactured from steel of high tensile strength.

The rivets, as in the case of the roller chain, take the load on the chain; they are, therefore, of the largest diameter possible relative to the breaking load of the bushed plates in which they work. They are manufactured from nickel steel rods, which are ground truly round and to size and hardened.

Bushes are inserted in the plates to provide a hardened bearing for the rivets. They are manufactured from mild steel, case-hardened, gauged for length, internal and external diameter, and hardened.

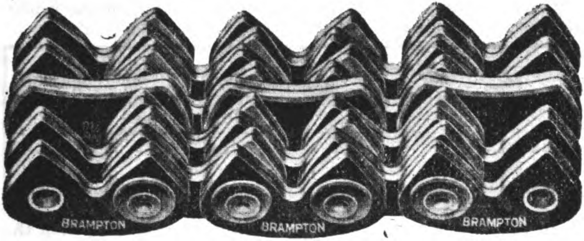


FIG. 22.—INVERTED TOOTH CHAIN.

The principle of the inverted tooth-type chain drive is that of a tooth-to-tooth engagement, the teeth of the chain engaging the wheel teeth over a considerable proportion of the wheel diameter.

DATA.

Gear ratio (maximum)—

For roller type chain	7 : 1
Inverted tooth type chain	6 : 1

Pinions should, if permissible, have an odd number of teeth, as this will give uniform wear on wheels and chain.

Number of teeth in wheels.	Roller chain.	Inverted tooth type.
Desirable maximum . .	17	19
,, maximum . .	120	120*

* This applies only to inverted tooth chains with a profile based on a 60° link face angle.

For chains with a profile based on a 75° link face angle, the maximum number of teeth in the wheel is 63.

Chain speed (feet per *minute*)—

Normal maximum, assuming the number of teeth be not less than 19 for roller type and 21 for inverted tooth type.

	Roller type.	Inverted tooth type.
$\frac{3}{8}''$ and $\frac{1}{2}''$ pitch . .	1000	1500
$\frac{5}{8}''$ and $\frac{3}{4}''$ pitch . .	900	1400
1″ and $1\frac{1}{4}''$ pitch . .	800	1350
$1\frac{1}{2}''$ and $1\frac{3}{4}''$ pitch . .	700	1300
2″ and $2\frac{1}{2}''$ pitch . .	600	1250
3″ pitch	500	1200

Centre distance between Shafts. Desirable maximum centre distance (expressed in pitches of chain).

Pitch of chain.	Type of chain.	
	Roller.	Inverted tooth
$\frac{3}{8}''$—$\frac{3}{4}''$	100	70
1″—$1\frac{3}{4}''$	85	60
2″ upwards	70	50

In the case of inclined drives with the wheel higher than the pinion, and with an inclination up to 60°, the horizontal component of the actual centre distance should not exceed the foregoing values.

In the case of vertical drives and those with an inclination approaching the vertical, the chain manufacturers should be consulted.

Desirable minimum. The centre distance should be such that the arc of contact between the chain and any wheel is not less than 120°, and with at least 7 teeth in engagement.

Adjustment of centres. Wherever possible some means should be provided for the adjustment of shaft centres, so that the slack in a worn chain may be taken up. The amount of the adjustment should not be less than $1\frac{1}{2}$ times the pitch of the chain.

Direction of rotation. A chain works best when running with the tight or driving side uppermost.

Normal maximum joint pressure per square inch of rivet wearing area—

(*a*) Roller type chain	2,000 lbs.
(*b*) Inverted tooth type chain	1,500 lbs.

FORMULAE FOR CHAIN WHEEL SIZES AND PROFILES.

Wheel for Roller Chain.

A = Pitch line dia.
B = Bottom of teeth dia. = Pitch line dia.—Roller dia.

$$C = \frac{3 \times P}{16}.$$

D = Top of teeth dia. = pitch line dia. + $\frac{\text{Pitch}}{2}$

R = Roller dia.
W = Width of roller—·02 pitch.

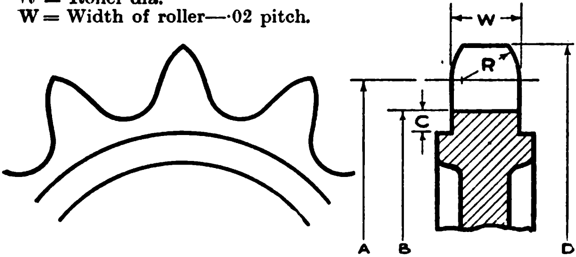


FIG. 23.—ROLLER CHAIN WHEEL.

Wheel for inverted tooth type chain.

A = Pitch line dia.

B = Top of teeth dia. = Pitch of chain × cotan $\left(\frac{180^\circ}{N}\right)$.

D = *See* table below.
R1 = 4P.

$R2 = \frac{P}{8}$. $R3 = \frac{P}{10}$.

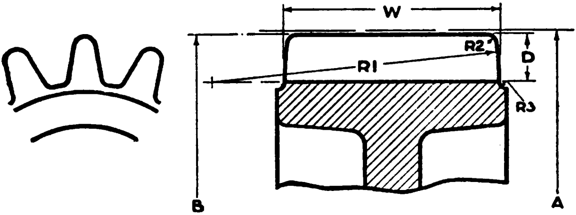


FIG. 24.—INVERTED TOOTH CHAIN WHEEL.

W = Tooth width. W(maximum) = 99 per cent of effective width of chain. W(minimum) = W(maximum) —Tolerance in table below.

Pitch . .	8mm	10 mm	½″	⅝″	¾″	1″	1¼″	1½″	1¾″
D . .	·200″	·259″	·325″	·403″	·500″	·650″	·803″	1·00″	1·14″
Tolerance .	·008″	·010	″·013″	·016″	·020″	·025″	·031″	·038″	·045″

ROPES AND ROPE DRIVING.

Speed of ropes vary from 3,000 to 6,000 feet per minute ; the latter speed should not be exceeded.

Cotton ropes vary from $\frac{3}{4}$ of an inch to $2\frac{1}{4}$ inches in diameter.

The stress on cotton, manilla, and hemp ropes ranges from 150 to 300 lbs. per square inch of the circumscribing circle ; the latter should not be exceeded by the best ropes.

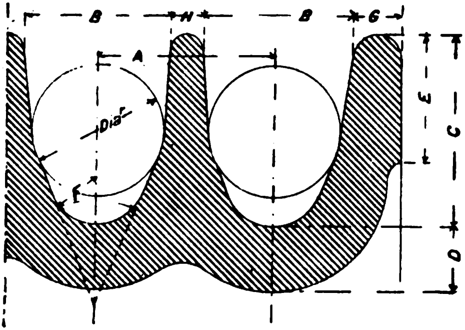


FIG. 25.—ROPE WHEELS.

TABLE 4.

(PROPORTIONS OF GROOVES FOR ROPE WHEELS. *Fig.* 25.)

Diameter of rope in inches.	A inches.	B ins.	C ins.	D ins.	E ins.	F ins.	G ins.	H ins.
$\frac{1}{2}$	1	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{5}{16}$	$\frac{1}{4}$
$\frac{5}{8}$	$1\frac{1}{8}$	$\frac{7}{8}$	1	$\frac{3}{8}$	$\frac{5}{8}$	$\frac{7}{32}$	$\frac{3}{8}$	$\frac{1}{4}$
$\frac{3}{4}$	$1\frac{5}{16}$	1	$1\frac{3}{16}$	$\frac{7}{16}$	$\frac{3}{4}$	$\frac{7}{32}$	$\frac{3}{8}$	$\frac{1}{4}$
$\frac{7}{8}$	$1\frac{7}{16}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{5}{16}$
1	$1\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{9}{16}$	$\frac{9}{16}$	1	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{5}{16}$
$1\frac{1}{4}$	$1\frac{7}{8}$	$1\frac{9}{16}$	$1\frac{15}{16}$	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{13}{32}$	$\frac{1}{2}$	$\frac{11}{32}$
$1\frac{1}{2}$	$2\frac{3}{16}$	$1\frac{13}{16}$	$2\frac{1}{4}$	$\frac{3}{4}$	$1\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{3}{8}$
$1\frac{5}{8}$	$2\frac{5}{16}$	$1\frac{15}{16}$	$2\frac{1}{2}$	$\frac{13}{16}$	$1\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{8}$
$1\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{16}$	$2\frac{5}{8}$	$\frac{7}{8}$	$1\frac{3}{4}$	$\frac{11}{16}$	$\frac{11}{16}$	$\frac{13}{32}$
2	$2\frac{3}{4}$	$2\frac{5}{16}$	3	1	2	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{16}$
$2\frac{1}{8}$	$2\frac{7}{8}$	$2\frac{7}{16}$	$3\frac{3}{16}$	$1\frac{1}{8}$	$2\frac{1}{4}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{15}{32}$

TABLE 5.—BRAKE HORSE POWER OF COTTON ROPES.

Speed in feet per minute.	Diameter of rope in inches.							
	¾-in.	⅞-in.	1-in.	1⅛-in.	1¼-in.	1½-in.	1¾-in.	2-in.
1500	3·75	5·5	6·75	8·5	10·5	15	21	27
2000	5	7	9	11·5	14	20·5	28	36
2250	5·5	7·75	10	12·5	16	23	31	41
2500	6	8·5	11	14	17	25	34	44
2750	6·5	9	11	15	18	26	36	47
3000	7	9·5	12	16	20	28	39	50
3250	7·5	10	13	16·5	20	29	40	53
3500	8	10·5	14	17·5	22	31	43	56
3750	8·25	11	14	18	22	32	44	57
4000	8·5	11·5	15	18·5	23	33	45	59
4250	8·75	11·5	15	19	24	34	46	60
4500	9	11·75	15·5	19	24	34	47	61
4750	9·5	12	15·5	19·25	24·5	35	47	61
5000	9·5	12	16	19·5	25	35·5	47·5	62
5500	8·5	11·5	14·8	18·75	23·5	33·5	45·5	59·5
6000	7·75	10·5	14	17·5	21·75	31·5	42·5	55·5

TABLE 6.—HORSE-POWER OF MANILLA ROPES.

Diameter of rope in inches.	Speed of rope in feet per minute.								
	1000	1500	2000	2500	3000	3500	4000	4500	5000
	Horse-power.								
¾	1¾	2¾	3½	4½	5½	6½	7	8¼	9
1	3½	4⅞	6½	8	10	11	13	15	16
1¼	5⅛	7½	10¼	13	15	18	20	23	26
1½	7½	11	15	18	22	26	30	34	37
1¾	10	15	20	25	30	35	40	45	50
2	13	19½	26	33	39	46	52	59	65

TABLE 7.—HORSE-POWER TRANSMITTED BY WIRE ROPES AT VARIOUS SPEEDS (CRADOCK).

Diameter of pulleys. Ft.	Diameter of rope. Ins.	Ratio of diameter of pulley to diameter of rope.	Rope speed in feet per second							
			20	30	40	50	60	70	80	90
4	$\frac{5}{16}$	153·6	6·5	8·7	11·0	13·3	15·6	18·0	20·0	22·0
	$\frac{3}{8}$	128·0	6·6	9·0	11·5	13·7	16·0	18·5	20·5	22·5
	$\frac{7}{16}$	110·0	6·8	9·4	12·0	14·2	16·5	19·0	21·0	23·0
5	$\frac{3}{8}$	160·0	8·8	12·4	16·0	19·5	23·0	26·5	29·0	32·0
	$\frac{7}{16}$	137·0	9·9	13·7	17·5	20·8	24·0	29·0	31·0	34·0
	$\frac{1}{2}$	120·0	12·0	17·0	22·0	26·4	30·5	35·0	39·0	43·0
6	$\frac{7}{16}$	165·0	11·6	16·3	21·0	25·5	29·5	34·0	38·0	42·0
	$\frac{1}{2}$	144·0	14·9	21·0	27·0	33·0	38·0	44·0	49·0	54·0
	$\frac{9}{16}$	128·0	16·0	24·0	31·0	37·0	42·0	48·0	53·0	58·5
7	$\frac{1}{2}$	168·0	17·0	23·5	30·0	37·0	43·0	49·5	55·0	60·5
	$\frac{9}{16}$	149·0	18·0	25·5	33·0	40·0	47·0	54·0	60·0	66·0
	$\frac{5}{8}$	134·0	22·4	31·7	41·0	50·0	58·0	67·0	75·0	82·5
8	$\frac{9}{16}$	170·0	22·5	31·3	40·0	48·5	56·0	64·5	72·0	79·0
	$\frac{5}{8}$	153·6	26·7	38·0	49·0	60·0	70·0	80·5	90·0	99·0
	$\frac{11}{16}$	140·0	30·4	44·0	58·0	71·0	83·0	96·0	107·0	117·5

TABLE 7 (*contd.*).—HORSE-POWER TRANSMITTED BY WIRE ROPES AT VARIOUS SPEEDS (CRADOCK).

Diameter of pulleys. Ft.	Diameter of rope. Ins.	Ratio of diameter of pulley to diameter of rope.	Rope speed in feet per second.							
			20	30	40	50	60	70	80	90
9	$\frac{5}{8}$	173·0	29·2	41·6	54·0	67·0	78·0	89·5	100·0	110·5
	$\frac{11}{16}$	157·0	33·4	47·5	62·0	76·0	89·0	103·0	115·0	126·5
	$\frac{3}{4}$	144·0	36·0	51·5	67·0	82·0	96·0	111·0	125·0	137·0
10	$\frac{11}{16}$	175·0	37·7	54·0	70·0	86·0	100·0	116·0	130·0	143·0
	$\frac{3}{4}$	160·0	39·2	55·6	72·0	87·5	102·0	117·5	132·0	145·0
	$\frac{13}{16}$	148·0	39·8	57·0	73·0	89·0	104·0	120·0	134·0	147·5
11	$\frac{3}{4}$	176·0	42·0	60·0	78·0	96·0	102·0	129·0	145·0	160·0
	$\frac{13}{16}$	162·0	44·4	64·0	83·0	102·0	119·0	138·0	155·0	171·0
	$\frac{7}{8}$	151·0	48·6	69·0	89·0	108·0	126·0	146·0	163·0	180·0
12	$\frac{3}{4}$	192·0	43·9	63·0	82·5	100·0	119·0	138·0	155·0	171·0
	$\frac{16}{13}$	177·0	46·3	66·0	86·3	106·0	124·0	144·0	161·0	177·0
	$\frac{7}{8}$	165·0	50·7	73·4	96·1	118·0	139·0	161·0	182 0	200·0
	$\frac{15}{16}$	153·6	53·5	76·5	99·5	122·0	143·0	166·0	185·0	204·0
	1	144·0	68·6	91·6	114·6	134·0	153·0	173·0	191·0	210·0

WIRE ROPES.

TABLE 8.—HORSE-POWER OF WIRE ROPE (Unwin).

Diameter of rope in inches	Diameter of pulleys in feet.	Number of revolutions per minute.	Breaking strength of rope in lbs.	Horse-power transmitted.	Velocity of belt in feet per second.
$\frac{7}{16}$	5	100	4260	8·6	26
$\frac{15}{32}$	6	100	5660	13·4	31
$\frac{1}{2}$	7	100	8200	21·1	36
$\frac{5}{8}$	8	100	11600	27·5	42
$\frac{5}{8}$	8	120	11600	33·0	50
$\frac{5}{8}$	9	100	11600	51·9	47
$\frac{5}{8}$	9	120	11600	62·2	56
$\frac{11}{16}$	10	100	15200	73·0	52
$\frac{11}{16}$	10	120	15200	87·6	62
$\frac{11}{16}$	10	140	15200	102·2	73
$\frac{11}{16}$	12	100	15200	116·7	63
$\frac{3}{4}$	12	120	17600	148·9	75
$\frac{3}{4}$	12	140	17600	173·7	87
$\frac{3}{4}$	14	100	17600	185·0	73
$\frac{3}{4}$	14	120	17600	222·0	87
$\frac{3}{4}$	15	120	17600	300·0	94

The ropes have each 42 wires.

BELTS AND BELTING.

BELT DATA.

Let T = tension on driving side in pounds.
t = tension on slack side in pounds.
θ = angle of lap of belt on pulley in radians.
μ = coefficient of friction.

Then $\frac{T}{t} = e^{\mu\theta}$

TABLE 9.—VALUES OF $e^{\mu\theta} = \frac{T}{t}$.

Angle of lap in deg.	Coefficient μ.								
	·25	·30	·35	·40	·45	·50	·55	·60	1·00
100	1·55	1·69	1·84	2·01	2·19	2·40	2·61	2·85	5·73
105	1·58	1·73	1·90	2·08	2·28	2·49	2·74	3·00	6·25
110	1·62	1·78	1·96	2·16	2·37	2·61	2·88	3·17	6·82
115	1·65	1·83	2·01	2·23	2·47	2·72	3·02	3·33	7·43
120	1·69	1·88	2·08	2·31	2·57	2·84	3·16	3·51	8·11
125	1·72	1·93	2·15	2·39	2·67	2·96	3·32	3·70	8·86
130	1·76	1·97	2·21	2·47	2·77	3·10	3·48	3·88	9·62
135	1·80	2·03	2·28	2·57	2·89	3·24	3·66	4·11	10·5
140	1·84	2·08	2·37	2·66	3·00	3·38	3·83	4·34	11·5
145	1·88	2·14	2·43	2·75	3·12	3·54	4·02	4·55	12·5
150	1·92	2·19	2·50	2·85	3·25	3·70	4·22	4·80	13·7
155	1·97	2·25	2·58	2·95	3·38	3·86	4·43	5·05	14·9
160	2·01	2·31	2·66	3·06	3·51	4·04	4·65	5·33	16·3
165	2·05	2·37	2·74	3·16	3·65	4·21	4·88	5·61	17·8
170	2·10	2·44	2·83	3·28	3·80	4·41	5·11	5·94	19·4
175	2·15	2·50	2·91	3·39	3·95	4·62	5·37	6·25	21·2
180	2·19	2·57	3·00	3·51	4·11	4·81	5·63	6·60	23·1
185	2·24	2·64	3·10	3·64	4·28	5·01	5·91	6·97	25·2
190	2·29	2·70	3·19	3·77	4·45	5·22	6·20	7·30	27·5
195	2·34	2·77	3·29	3·90	4·62	5·50	6·50	7·69	30·0
200	2·39	2·85	3·39	4·04	4·81	5·70	6·82	8·10	32·8
205	2·45	2·93	3·50	4·18	5·00	6·00	7·16	8·55	35·7
210	2·50	3·00	3·61	4·33	5·20	6·28	7·51	9·00	39·0
215	2·56	3·08	3·72	4·49	5·41	6·50	7·88	9·49	42·5
220	2·61	3·16	3·83	4·63	5·63	6·82	8·27	10·0	46·5
225	2·67	3·25	3·95	4·81	5·85	7·13	8·67	10·6	50·7
230	2·73	3·33	4·07	4·97	6·07	7·43	9·07	11·1	55·0
235	2·79	3·42	4·20	5·16	6·33	7·78	9·55	11·7	60·4
240	2·85	3·51	4·33	5·34	6·58	8·13	10·02	12·3	66·0

TABLE 10.—APPROXIMATE HORSE-POWER TRANSMITTED BY BELTS.

Double Leather Belts.

Speed in feet per minute.	Width of belt in inches.										
	4	6	8	10	12	14	16	18	20	22	24
	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
400	2¾	4½	5¾	7¼	8½	10	11½	14	14½	16	17½
600	4¼	6½	8¾	11	13	15	17½	19½	22	24	26
800	5¾	8½	11½	14½	17½	20½	23	26	29	32	34½
1000	7¼	11	14½	18¼	21½	25½	29	32½	36	40	41½
1200	8½	13	17½	22	26	30½	34½	39	44	48	52½
1500	10¾	16¼	21¾	27¼	32½	38	43½	49	54½	60	65½
1800	13	19½	26	32¾	39	45½	52	59	65½	72	78½
2000	14½	21¾	29	36¼	43½	50½	58	65½	72½	80	87
2400	17½	26	34¾	44	52½	60½	69½	78½	88	96	105
2800	20¼	30½	40½	51	61	71	81	91½	102	112	122
3000	21½	32½	43½	54½	65½	76	87½	98	108	120	131
3500	25½	38	50¾	63½	76	89	101	114	127	140	153
4000	29	43½	58¼	72¾	87	101	116	131	145	160	174
4500	32½	49	65	82	98	114	131	147	163	180	196
5000	36½	54½	72¾	91	109	127	147	163	182	200	218

TABLE 11.—DRIVING POWER OF LEATHER BELTS (F. A. Halsey).

Diameter of pulley in inches.	Revolutions of pulley per minute.											
	50	60	70	80	90	100	125	150	175	200	250	300
	Horse-power transmitted by each inch width of single belt.											
12	·20	·24	·28	·32	·35	·39	·49	·59	·69	·79	·98	1·18
14	·23	·28	·32	·37	·41	·46	·57	·69	·80	·92	1·14	1·38
16	·26	·32	·37	·42	·47	·53	·66	·79	·92	1·05	1·31	1·58
18	·30	·35	·41	·47	·53	·59	·74	·89	1·04	1·18	1·48	1·77
20	·33	·39	·46	·52	·59	·65	·82	·98	1·14	1·31	1·64	1·96
24	·39	·47	·55	·63	·71	·79	·98	1·18	1·38	1·58	1·97	2·36
28	·46	·55	·64	·73	·83	·92	1·15	1·38	1·61	1·84	2·29	2·75
32	·52	·63	·73	·84	·94	1·05	1·31	1·57	1·84	2·10	2·63	3·15
36	·59	·71	·83	·95	1·06	1·18	1·48	1·77	2·06	2·36	2·96	3·55
40	·65	·79	·92	1·04	1·18	1·31	1·64	1·96	2·29	2·62	3·27	3·94
44	·72	·87	1·01	1·15	1·30	1·45	1·80	2·16	2·53	2·89	3·60	4·34
48	·79	·94	1·10	1·26	1·42	1·57	1·96	2·36	2·75	3·15	3·93	4·72
54	·89	1·06	1·24	1·42	1·59	1·77	2·22	2·66	3·10	3·55	4·43	5·31
60	·98	1·18	1·38	1·57	1·77	1·96	2·46	2·95	3·44	3·93	4·91	5·90
66	1·08	1·30	1·52	1·73	1·95	2·17	2·71	3·25	3·79	4·33	5·41	6·50
72	1·18	1·42	1·65	1·89	2·13	2·36	2·95	3·54	4·13	4·72	5·90	7·10
78	1·28	1·53	1·79	2·04	2·30	2·56	3·20	3·84	4·48	5·11	6·40	7·68
84	1·38	1·65	1·93	2·21	2·48	2·75	3·44	4·13	4·81	5·51	6·89	8·28
90	1·48	1·77	2·06	2·36	2·66	2·96	3·68	4·43	5·18	5·90	7·38	8·85
96	1·57	1·89	2·21	2·53	2·84	3·15	3·95	4·73	5·51	6·31	7·89	9·15

TABLE 12.—HORSE POWER PER INCH OF WIDTH OF LEATHER BELT ALLOWING FOR CENTRIFUGAL FORCE

$\mu = 0{\cdot}4$. Angle of Lap, 180 Degrees.

Speed in ft. per min.	Single belt, $\frac{3}{16}$ in. thick.		Double belt, $\frac{3}{8}$ in. thick.	
	Cemented max. stress 400 lb./sq. in.	Laced max. stress 320 lb./sq. in.	Cemented max. stress 300 lb./sq. in.	Laced max. stress 240 lb./sq. in
100	·16	·13	·24	·19
200	·32	·26	·48	·38
300	·48	·38	·72	·58
400	·64	·51	·96	·77
500	·80	·64	1·20	·96
600	·96	·77	1·44	1·15
700	1·12	·90	1·68	1·34
800	1·28	1·02	1·92	1·54
900	1·44	1·15	2·16	1·73
1000	1·60	1·28	2·40	1·92
1100	1·75	1·40	2·62	2·09
1200	1·90	1·52	2·84	2·27
1300	2·06	1·65	3·05	2·44
1400	2·21	1·77	3·27	2·62
1500	2·36	1·89	3·49	2·79
1600	2·51	2·01	3·71	2·96
1700	2·66	2·13	3·93	3·14
1800	2·82	2·26	4·14	3·31
1900	2·97	2·38	4·36	3·49
2000	3·12	2·50	4·58	3·66
2200	3·37	2·68	4·93	3·91
2400	3·63	2·87	5·29	4·16
2600	3·89	3·05	5·64	4·40
2800	4·14	3·24	6·00	4·65
3000	4·40	3·42	6·35	4·90
3200	4·62	3·58	6·63	5·08
3400	4·83	3·73	6·89	5·24
3600	5·01	3·84	7·11	5·36
3800	5·20	3·96	7·30	5·46
4000	5·36	4·06	7·47	5·52
4200	5·51	4·14	7·60	5·56
4400	5·64	4·21	7·70	5·57
4600	5·75	4·26	7·76	5·53
4800	5·84	4·29	7·79	5·46
5000	5·90	4·28	7·75	5·32
5200	5·95	4·26	7·68	5·14
5400	5·98	4·23	7·58	4·95
5600	5·98	4·16	7·41	4·68
5800	5·95	4·07	7·20	4·38
6000	5·92	3·97	6·96	4·04

The values given in Table 12 should be multiplied by the following factors for angles of lap other than 180°, the value of μ remaining 0·4.

Angle of lap, degrees	100	110	120	130	140	150	160	170
Multiplying factor.	0·7	0·75	0·79	0·83	0·87	0·91	0·94	0·97

Angle of lap, degrees	180	190	200	210	220	230	240
Multiplying factor	1·00	1·03	1·05	1·08	1·10	1·12	1·14

For values of μ and angle of lap other than 0·4 and 180°, multiply the values given in Table 12 by the factor $1{\cdot}4\,(e^{\mu\theta} - 1)/e^{\mu\theta}$, $e^{\mu\theta}$ being read from Table 9.

Horse-power transmitted—

Let P = effective driving load per inch width.
D = diameter of pulley, in inches.
W = width of belt, in inches.
N = revolutions per minute.
V = velocity of belt, in feet per min.—
$= \pi$ DN/12.

Then horse-power—

$$= PWV/33000 = PW\pi DN/12 \times 33000$$
$$= PWDN/12600.$$

Average maximum permissible stress.	Lbs. per sq. in.
Single belts (cemented) . . .	400
ditto (laced) . . .	320
Double belts (cemented) . . .	300
ditto (laced) . . .	240

LENGTH OF BELTS.

Using the construction shown in Fig. 26, the length L of the open belt is given by L = 2 (AC + CD + DE).

Then OF = CD = S cos θ.

If θ be expressed in radians, AC = $\frac{D}{2}\left(\frac{\pi}{2}+\theta\right)$ and DE = $\frac{d}{2}\left(\frac{\pi}{2}-\theta\right)$.

$$\text{Whence } L = 2\left\{\frac{D}{2}\left(\frac{\pi}{2}+\theta\right) + S.\cos\theta + \frac{d}{2}\left(\frac{\pi}{2}-\theta\right)\right\}$$

$$\text{or } L = \left\{\frac{\pi}{2}(D+d) + (D-d)\,\theta + 2\,S.\cos\theta\right\}.$$

Also, sin θ = (D − d) /2S.

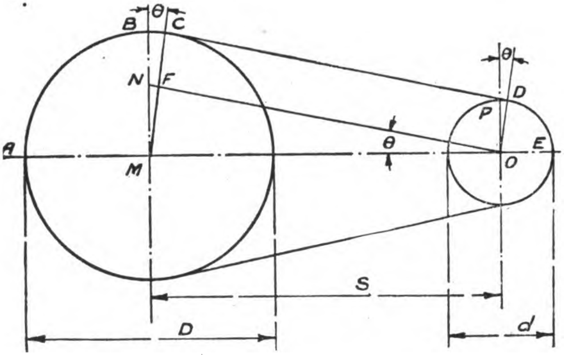


FIG. 26.—LENGTH OF OPEN BELT.

The angle of lap in radians is ($\pi + 2\theta$) for the larger pulley, and ($\pi - 2\,\theta$) for the smaller pulley.

Approximate length of open belt.

$$L = \left\{\frac{\pi}{2}\,.\,(D+d) + \sqrt{\frac{(D-d)^2}{4} + S^2}\right\}.$$

Length of crossed belt. Referring to Fig. 27, the length L of the crossed belt is given by L = 2{AC + CD + DF}.

$$\text{Then } L = 2\left\{\frac{D}{2}\cdot\left(\frac{\pi}{2}+\theta\right) + S.\cos\theta + \frac{d}{2}\cdot\left(\frac{\pi}{2}+\theta\right)\right\}$$

$$\text{or } L = \left\{\left(\frac{\pi}{2}+\theta\right)(D+d) + S.\cos\theta\right\}.$$

$$\text{Now the arc } KH = \left(\frac{\pi}{2}+\theta\right)(D+d)$$

$$\therefore L = 2\ \{KH + OH\}$$

L is therefore constant when $(D + d)$ and S are constant.

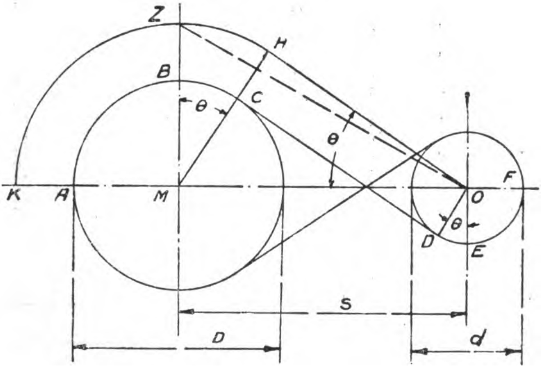


FIG. 27.—LENGTH OF CROSSED BELT.

Approximate length of crossed belt.

$$L = \frac{\pi}{2}(D + d) + 2\sqrt{S^2 + \frac{(D+d)^2}{4}}.$$

ROPES AND ROPE FITTINGS.

TABLE 13.—WEIGHT AND STRENGTH OF ROUND ROPES.

Hemp Ropes.		Steel Wire Ropes.		Iron Wire Ropes.		Equivalent Strength.	
Circumference in inches.	Weight per fathom in lbs.	Circumference in inches.	Weight per fathom in lbs.	Circumference in inches.	Weight per fathom in lbs.	Working Load Cwts.	Breaking Load Tons.
2¾	2	1	1			6	2
		1½	1½	1	1	9	3
3¾	4	1⅝	2			12	4
		1¾	2½	1½	1½	15	5
4½	5	1⅞	3			18	6
		2	3½	1⅝	2	21	7
5½	7	2⅛	4	1¾	2½	24	8
		2¼	4½			27	9
6	9	2⅜	5	1⅞	3	30	10
		2½	5½			33	11
6½	10	2⅝	6	2	3½	36	12
		2¾	6½	2⅛	4	39	13
7	12	2⅞	7	2¼	4½	42	14
		3	7½			45	15
7½	14	3⅛	8	2⅜	5	48	16
		3¼	8½			51	17
8	16	3⅜	9	2½	5½	54	18
		3½	10	2⅝	6	60	20
8½	18	3⅝	11	2¾	6½	66	22
		3¾	12			72	24
9½	22	3⅞	13	3¼	8	78	26
10	26	4	14			84	28
		4½	15	3⅜	9	90	30
11	30	4⅜	16			96	32
		4½	18	3½	10	108	36
12	34	4⅝	20	3¾	12	120	40

TABLE 14.—WEIGHT AND STRENGTH OF FLAT ROPES.

Hemp Ropes.		Steel Wire Ropes.		Iron Wire Ropes.		Equivalent Strength.	
Size in inches.	Weight per fathom in lbs.	Size in inches.	Weight per fathom in lbs.	Size in inches.	Weight per fathom in lbs.	Working Load Cwts.	Breaking Load Tons.
4 × 1⅓	20			2¼ × ½	11	44	20
5 × 1¾	24			2½ × ½	13	52	23
5½ × 1⅜	26			2¾ × ⅝	15	60	27
5¾ × 1½	28	2 × ½	10	3 × ⅝	16	64	28
6 × 1½	30	2¼ × ½	11	3¼ × ⅝	18	72	32
7 × 1⅞	36	2¼ × ½	12	3½ × ⅝	20	80	36
8¼ × 2⅛	40	2½ × ½	13	3¾ × 11/16	22	88	40
8½ × 2¼	45	2¾ × ⅜	15	4 × 11/16	25	100	45
9 × 2½	50	3 × ⅜	16	4¼ × ¾	28	112	50
9½ × 2⅜	55	3¼ × ⅜	18	4½ × ¾	32	128	56
10 × 2½	60	3½ × ⅜	20	4⅝ × ¾	34	136	60

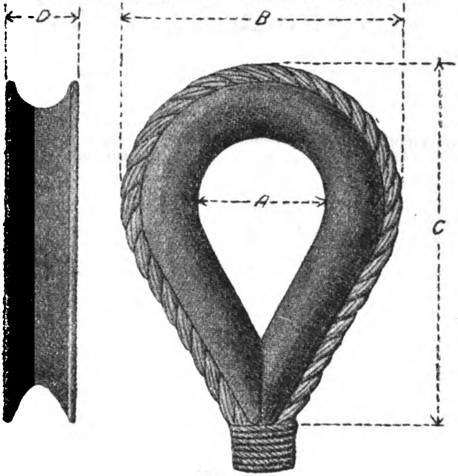


FIG. 28.

TABLE 15.—STANDARD THIMBLES. (*Fig.* 28.)

Diameter of Rope.	A	B	C	D
Inches.	Inches.	Inches.	Inches.	Inches.
$\frac{5}{16}$	$\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$\frac{1}{2}$
$\frac{3}{8}$	$\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{3}{4}$	$\frac{5}{8}$
$\frac{7}{16}$	$\frac{7}{8}$	$2\frac{1}{4}$	$2\frac{3}{4}$	$\frac{5}{8}$
$\frac{1}{2}$	$1\frac{1}{8}$	$2\frac{5}{8}$	$2\frac{3}{4}$	$\frac{11}{16}$
$\frac{9}{16}$	$1\frac{1}{8}$	$2\frac{3}{4}$	$2\frac{3}{4}$	$\frac{11}{16}$
$\frac{5}{8}$	$1\frac{5}{16}$	$3\frac{1}{8}$	$3\frac{1}{8}$	$\frac{11}{16}$
$\frac{11}{16}$	$1\frac{5}{16}$	$3\frac{1}{4}$	$3\frac{1}{8}$	$\frac{11}{16}$
$\frac{3}{4}$	$2\frac{1}{8}$	$4\frac{1}{8}$	$4\frac{1}{4}$	1
$\frac{13}{16}$	$2\frac{1}{8}$	$4\frac{1}{4}$	$4\frac{1}{4}$	1
$\frac{7}{8}$	$2\frac{1}{8}$	5	$5\frac{3}{4}$	$1\frac{1}{4}$
$\frac{15}{16}$	$2\frac{1}{8}$	$5\frac{1}{8}$	$5\frac{3}{4}$	$1\frac{1}{4}$
1	$2\frac{1}{2}$	$5\frac{1}{2}$	6	$1\frac{1}{2}$
$1\frac{1}{16}$	$2\frac{1}{2}$	$5\frac{5}{8}$	6	$1\frac{1}{2}$
$1\frac{1}{8}$	$2\frac{7}{8}$	6	8	$1\frac{3}{4}$
$1\frac{1}{4}$	$2\frac{7}{8}$	$6\frac{3}{8}$	8	$1\frac{3}{4}$
$1\frac{3}{8}$	$3\frac{3}{8}$	$7\frac{3}{8}$	$9\frac{1}{2}$	$2\frac{1}{8}$
$1\frac{1}{2}$	$3\frac{3}{8}$	$7\frac{5}{8}$	$9\frac{1}{2}$	$2\frac{1}{8}$
$1\frac{5}{8}$	4	$8\frac{1}{2}$	$10\frac{3}{4}$	$2\frac{3}{8}$
$1\frac{3}{4}$	4	$8\frac{3}{4}$	$10\frac{3}{4}$	$2\frac{3}{8}$
2	4	$9\frac{1}{2}$	$13\frac{1}{4}$	$2\frac{1}{2}$

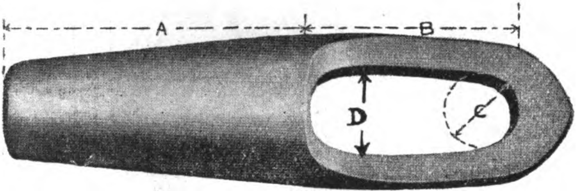


FIG. 29.

TABLE 16.—CLOSED CONICAL SOCKET.
(*Hand Forged.*) *Iron.*

Diameter of rope.	Length of barrel. A	Length of Eye. B.	Diameter admitted in eye. C.	Width of eye. D.
Inch.	Inches.	Inches.	Inches.	Inches.
$\frac{3}{8}$	$2\frac{5}{8}$	$1\frac{7}{8}$	$\frac{5}{8}$	$\frac{3}{4}$
$\frac{7}{16}$	$2\frac{5}{8}$	$1\frac{7}{8}$	$\frac{5}{8}$	$\frac{3}{4}$
$\frac{1}{2}$	$3\frac{3}{8}$	$2\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$
$\frac{9}{16}$	$3\frac{3}{8}$	$2\frac{1}{4}$	$\frac{3}{4}$	$\frac{7}{8}$
$\frac{5}{8}$	$3\frac{3}{4}$	$2\frac{11}{16}$	$\frac{7}{8}$	1
$\frac{3}{4}$	$4\frac{1}{2}$	3	1	$1\frac{1}{8}$
$\frac{7}{8}$	$5\frac{1}{4}$	$3\frac{7}{16}$	$1\frac{1}{8}$	$1\frac{1}{4}$
1	6	$3\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$
$1\frac{1}{8}$	$6\frac{3}{4}$	$4\frac{3}{16}$	$1\frac{3}{8}$	$1\frac{5}{8}$
$1\frac{1}{4}$	$7\frac{1}{2}$	$4\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{3}{4}$
$1\frac{3}{8}$	$8\frac{1}{4}$	$4\frac{15}{16}$	$1\frac{5}{8}$	$1\frac{7}{8}$
$1\frac{1}{2}$	9	$5\frac{3}{4}$	$1\frac{3}{4}$	2
$1\frac{5}{8}$	$9\frac{3}{4}$	$6\frac{3}{16}$	$1\frac{7}{8}$	$2\frac{1}{4}$
$1\frac{3}{4}$	$10\frac{1}{2}$	$6\frac{1}{2}$	2	$2\frac{1}{2}$
$1\frac{7}{8}$	$11\frac{1}{4}$	$6\frac{15}{16}$	$2\frac{1}{8}$	$2\frac{5}{8}$
2	12	$7\frac{3}{8}$	$2\frac{1}{4}$	$2\frac{7}{8}$

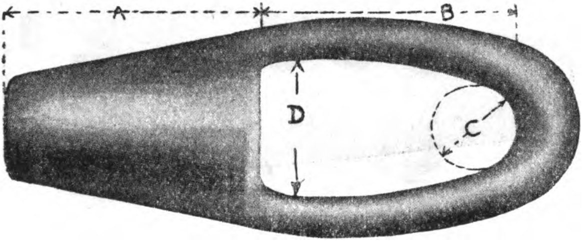


FIG. 30.

TABLE 17.—CLOSED CONICAL SOCKET.
(*Drop Forged.*) *Steel.*

Diameter of rope.	Length of barrel. A	Length of Eye. B.	Diameter admitted in eye. C.	Width of eye. D.
Inch.	Inches.	Inches.	Inches.	Inches.
1/4	1 1/2	1 5/8	1/2	9/16
3/8	2 1/4	2	5/8	5/8
7/16	2 1/4	2	5/8	5/8
1/2	2 1/2	2 1/4	7/8	3/4
9/16	2 1/2	2 1/4	7/8	3/4
5/8	2 3/4	2 3/4	1	7/8
3/4	3	3	1 1/4	1
7/8	3 1/2	3 1/2	1 1/2	1 1/8
1	4	4	1 5/8	1 3/8
1 1/8	4 1/2	4 1/2	1 3/4	1 1/2
1 1/4	5	5	1 7/8	1 3/4
1 3/8	5 1/2	5 1/2	2	2 1/8
1 1/2	5 1/2	5 1/2	2	2 1/8

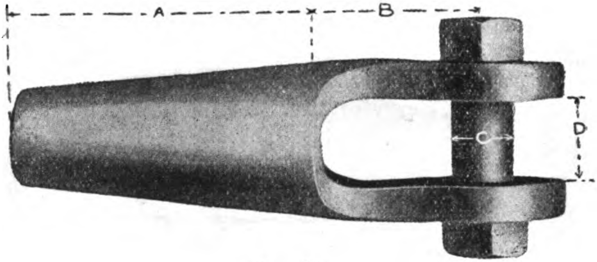


Fig. 31.

Table 18.—Open Conical Socket.
(Hand Forged.) Iron.
With Bolt and Nut.

Diameter of rope.	Length of barrel. A.	Centre of bolt to top of barrel. B.	Diameter of bolt. C.	Width between jaws. D.
Inch.	Inches.	Inches.	Inches.	Inches.
$\frac{3}{8}$	$2\frac{5}{8}$	$1\frac{9}{16}$	$\frac{5}{8}$	$\frac{7}{8}$
$\frac{7}{16}$	$2\frac{5}{8}$	$1\frac{9}{16}$	$\frac{5}{8}$	$\frac{7}{8}$
$\frac{1}{2}$	$3\frac{3}{8}$	$1\frac{7}{8}$	$\frac{3}{4}$	1
$\frac{9}{16}$	$3\frac{3}{8}$	$1\frac{7}{8}$	$\frac{3}{4}$	1
$\frac{5}{8}$	$3\frac{3}{4}$	$2\frac{1}{4}$	$\frac{7}{8}$	$1\frac{1}{8}$
$\frac{3}{4}$	$4\frac{1}{2}$	$2\frac{1}{2}$	1	$1\frac{1}{4}$
$\frac{7}{8}$	$5\frac{1}{4}$	$2\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$
1	6	$3\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{5}{8}$
$1\frac{1}{8}$	$6\frac{3}{4}$	$3\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{3}{4}$
$1\frac{1}{4}$	$7\frac{1}{2}$	$3\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{7}{8}$
$1\frac{3}{8}$	$8\frac{1}{4}$	$4\frac{1}{8}$	$1\frac{5}{8}$	2
$1\frac{1}{2}$	9	$4\frac{7}{8}$	$1\frac{3}{4}$	$2\frac{1}{8}$
$1\frac{5}{8}$	$9\frac{3}{4}$	$5\frac{1}{4}$	$1\frac{7}{8}$	$2\frac{3}{8}$
$1\frac{3}{4}$	$10\frac{1}{2}$	$5\frac{1}{2}$	2	$2\frac{5}{8}$
$1\frac{7}{8}$	$11\frac{1}{4}$	$5\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{3}{4}$
2	12	$6\frac{1}{4}$	$2\frac{1}{4}$	3

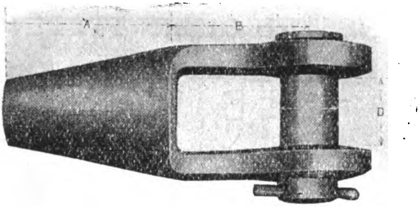


FIG. 32.

TABLE 19.—OPEN CONICAL SOCKET.

(Drop Forged.) Steel.
With Bolt and Split Pin.

Diameter of rope.	Length of barrel. A.	Centre of bolt to top of barrel. B.	Diameter of bolt. C.	Width of eye. D.
Inch.	Inches.	Inches.	Inches.	Inches.
$\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{5}{8}$
$\frac{3}{8}$	$2\frac{1}{4}$	$1\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{8}$
$\frac{7}{16}$	$2\frac{1}{4}$	$1\frac{3}{4}$	$\frac{5}{8}$	$1\frac{1}{16}$
$\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{7}{8}$	$\frac{3}{4}$	$1\frac{1}{16}$
$\frac{9}{16}$	$2\frac{1}{2}$	$1\frac{7}{8}$	$\frac{3}{4}$	$1\frac{1}{16}$
$\frac{5}{8}$	$2\frac{3}{4}$	$2\frac{1}{8}$	$\frac{7}{8}$	$1\frac{1}{8}$
$\frac{3}{4}$	3	$2\frac{1}{2}$	1	$1\frac{1}{4}$
$\frac{7}{8}$	$3\frac{1}{2}$	$2\frac{7}{8}$	$1\frac{1}{4}$	$1\frac{7}{16}$
1	4	$3\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{5}{8}$
$1\frac{1}{8}$	$4\frac{1}{2}$	$3\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{7}{8}$
$1\frac{1}{4}$	5	$4\frac{1}{4}$	$1\frac{7}{8}$	$2\frac{1}{8}$
$1\frac{3}{8}$	$5\frac{1}{2}$	$4\frac{1}{2}$	2	$2\frac{1}{2}$
$1\frac{1}{2}$	$5\frac{1}{2}$	$4\frac{1}{2}$	2	$2\frac{1}{2}$

CHAINS AND CHAIN FITTINGS.

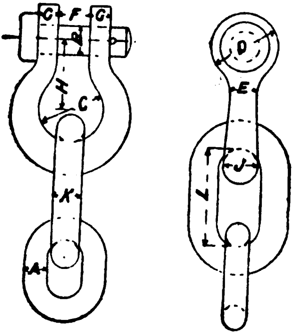


FIG. 33.

TABLE 20.—PROPORTIONS OF SHACKLES. (*Fig.* 33.)

Diameter of chain A inches.	B ins	C ins.	D ins.	E ins.	F ins.	G ins.	H ins.	J ins.	K ins.	L ins.
$\frac{3}{8}$	$\frac{1}{2}$	1	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{5}{16}$	1	$\frac{9}{16}$	$\frac{7}{16}$	$1\frac{5}{8}$
$\frac{7}{16}$	$\frac{9}{16}$	$1\frac{1}{8}$	1	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{3}{8}$	$1\frac{1}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$1\frac{3}{4}$
$\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{7}{16}$	$1\frac{1}{4}$	$\frac{11}{16}$	$\frac{9}{16}$	$1\frac{7}{8}$
$\frac{9}{16}$	$\frac{11}{16}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{1}{2}$	$1\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	2
$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{9}{16}$	$1\frac{1}{2}$	$\frac{7}{8}$	$\frac{11}{16}$	$2\frac{3}{8}$
$\frac{11}{16}$	$\frac{13}{16}$	$1\frac{3}{4}$	$1\frac{9}{16}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{5}{8}$	$1\frac{3}{4}$	1	$\frac{13}{16}$	$2\frac{5}{8}$
$\frac{3}{4}$	$\frac{7}{8}$	2	$1\frac{3}{4}$	$\frac{7}{8}$	1	$\frac{11}{16}$	2	$1\frac{1}{8}$	$\frac{7}{8}$	$2\frac{7}{8}$
$\frac{13}{16}$	$\frac{15}{16}$	$2\frac{1}{8}$	$1\frac{13}{16}$	$\frac{15}{16}$	$1\frac{1}{16}$	$\frac{3}{4}$	$2\frac{1}{8}$	$1\frac{3}{16}$	$\frac{15}{16}$	3
$\frac{7}{8}$	1	$2\frac{1}{4}$	$1\frac{15}{16}$	1	$1\frac{1}{8}$	$\frac{3}{4}$	$2\frac{1}{4}$	$1\frac{1}{4}$	1	$3\frac{1}{4}$
1	$1\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{4}$	$\frac{13}{16}$	$2\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$	$3\frac{5}{8}$
$1\frac{1}{8}$	$1\frac{3}{8}$	$2\frac{3}{4}$	$2\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	$3\frac{7}{8}$
$1\frac{1}{4}$	$1\frac{1}{2}$	3	$2\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{2}$	1	3	$1\frac{5}{8}$	$1\frac{3}{8}$	$4\frac{1}{4}$
$1\frac{3}{8}$	$1\frac{5}{8}$	$3\frac{1}{4}$	$2\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{1}{8}$	$3\frac{3}{8}$	$1\frac{13}{16}$	$1\frac{1}{2}$	$4\frac{5}{8}$
$1\frac{1}{2}$	$1\frac{3}{4}$	$3\frac{1}{2}$	3	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{3}{4}$	2	$1\frac{5}{8}$	5

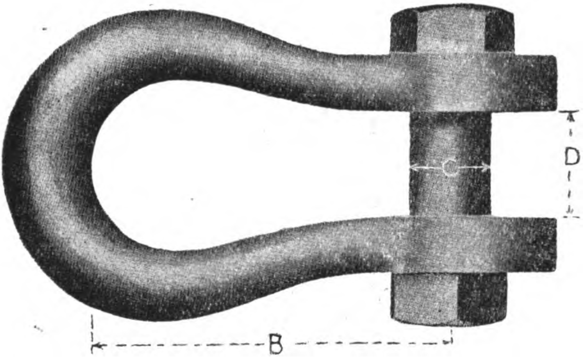


FIG. 34.

TABLE 21.—SHACKLE OR D LINK. (*Fig.* 34.)
(*Hand Forged.*) *Iron.*

No.	Centre of bolt to inside. B.	Diameter of bolt. C.	Width between jaws. D.	Breaking strain
	Inches.	Inches.	Inches.	Tons.
1	$1\frac{5}{16}$	$\frac{3}{8}$	$\frac{5}{8}$	4·8
2	$1\frac{3}{4}$	$\frac{1}{2}$	$\frac{3}{4}$	6·6
3	$2\frac{1}{16}$	$\frac{5}{8}$	$\frac{7}{8}$	8·6
4	$2\frac{5}{8}$	$\frac{3}{4}$	1	13·4
5	$3\frac{1}{16}$	$\frac{7}{8}$	$1\frac{1}{8}$	19·4
6	$3\frac{1}{2}$	1	$1\frac{1}{4}$	26·4
7	$3\frac{15}{16}$	$1\frac{1}{8}$	$1\frac{3}{8}$	34·4
8	$4\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{5}{8}$	43·6
9	$4\frac{13}{16}$	$1\frac{3}{8}$	$1\frac{3}{4}$	53·8
10	$5\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{7}{8}$	65·2
11	$5\frac{11}{16}$	$1\frac{5}{8}$	2	77·6
12	$6\frac{1}{8}$	$1\frac{3}{4}$	$2\frac{1}{8}$	91·2
13	$6\frac{9}{16}$	$1\frac{7}{8}$	$2\frac{3}{8}$	105·8
14	7	2	$2\frac{5}{8}$	121·4
15	$7\frac{7}{16}$	$2\frac{1}{8}$	$2\frac{3}{4}$	138·2
16	$7\frac{7}{8}$	$2\frac{1}{4}$	3	156·0

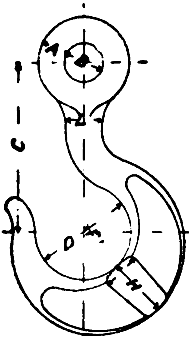


FIG. 35.—CRANE HOOKS.

TABLE 22.—PROPORTIONS OF CRANE-HOOKS.

Safe load in tons.	A inches.	B inches.	C inches.	D inches.	E inches.	F inch's.	G inches.	H inches.
$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$3\frac{13}{16}$	$2\frac{3}{8}$	1	$\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{1}{4}$
$\frac{3}{4}$	$\frac{13}{16}$	$\frac{13}{16}$	$4\frac{1}{8}$	$2\frac{1}{2}$	$1\frac{1}{16}$	$\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{7}{16}$
1	$\frac{7}{8}$	$\frac{7}{8}$	$4\frac{1}{2}$	$2\frac{11}{16}$	$1\frac{1}{8}$	$\frac{5}{16}$	$1\frac{5}{16}$	$1\frac{5}{8}$
$1\frac{1}{2}$	$\frac{7}{8}$	1	$4\frac{7}{8}$	$2\frac{13}{16}$	$1\frac{3}{16}$	$\frac{3}{8}$	$1\frac{5}{16}$	$1\frac{3}{4}$
2	$\frac{15}{16}$	$1\frac{1}{16}$	$5\frac{1}{4}$	3	$1\frac{1}{4}$	$\frac{7}{16}$	$1\frac{3}{8}$	2
3	1	$1\frac{1}{8}$	$5\frac{5}{8}$	$3\frac{1}{8}$	$1\frac{5}{16}$	$\frac{1}{2}$	$1\frac{3}{8}$	$2\frac{1}{8}$
4	1	$1\frac{3}{16}$	$5\frac{15}{16}$	$3\frac{5}{16}$	$1\frac{3}{8}$	$\frac{9}{16}$	$1\frac{7}{16}$	$2\frac{5}{16}$
5	$1\frac{1}{16}$	$1\frac{1}{4}$	$6\frac{5}{16}$	$3\frac{7}{16}$	$1\frac{7}{16}$	$\frac{5}{8}$	$1\frac{1}{2}$	$2\frac{1}{2}$
6	$1\frac{1}{8}$	$1\frac{3}{8}$	$6\frac{21}{32}$	$3\frac{5}{8}$	$1\frac{1}{2}$	$\frac{11}{16}$	$1\frac{1}{2}$	$2\frac{11}{16}$
8	$1\frac{1}{4}$	$1\frac{1}{2}$	$7\frac{3}{8}$	$3\frac{15}{16}$	$1\frac{5}{8}$	$\frac{11}{16}$	$1\frac{5}{8}$	3
10	$1\frac{3}{8}$	$1\frac{5}{8}$	$8\frac{1}{8}$	$4\frac{1}{4}$	$1\frac{3}{4}$	$\frac{3}{4}$	$1\frac{3}{4}$	$3\frac{5}{16}$
12	$1\frac{7}{16}$	$1\frac{13}{16}$	$8\frac{27}{32}$	$4\frac{9}{16}$	$1\frac{7}{8}$	$\frac{3}{4}$	$1\frac{7}{8}$	$3\frac{5}{8}$
15	$1\frac{9}{16}$	$1\frac{15}{16}$	$9\frac{9}{16}$	$4\frac{7}{8}$	2	$\frac{13}{16}$	2	$3\frac{7}{8}$
18	$1\frac{5}{8}$	$2\frac{1}{8}$	$10\frac{1}{4}$	$5\frac{3}{16}$	$2\frac{1}{8}$	$\frac{13}{16}$	$2\frac{1}{8}$	$4\frac{3}{16}$
21	$1\frac{3}{4}$	$2\frac{1}{4}$	11	$5\frac{1}{2}$	$2\frac{1}{4}$	$\frac{7}{8}$	$2\frac{1}{4}$	$4\frac{1}{2}$

TABLE 23.—PROPORTIONS, WEIGHT, AND STRENGTH OF ORDINARY CHAINS.

Diameter of Chain, inches.	Outside Length, inches.	Outside Width, inches.	Diagonal Width, inches.	Safe Load,		Test Load,		Weight per Fathom in lbs.
				Tons	cwts.	Tons	cwts.	
$\frac{1}{4}$	1	$\frac{7}{8}$	$\frac{11}{16}$		$12\frac{1}{2}$	1	$5\frac{1}{2}$	$3\frac{1}{2}$
$\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	$\frac{27}{32}$		$12\frac{1}{2}$	1	$5\frac{1}{2}$	$5\frac{1}{2}$
$\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1		18	1	17	8
$\frac{7}{16}$	$1\frac{3}{4}$	$1\frac{7}{16}$	$1\frac{5}{32}$	1	$4\frac{1}{2}$	2	10	$10\frac{1}{2}$
$\frac{1}{2}$	$2\frac{3}{8}$	$1\frac{11}{16}$	$1\frac{3}{8}$	1	12	3	5	14
$\frac{9}{16}$	$2\frac{11}{16}$	$1\frac{7}{8}$	$1\frac{1}{2}$	2	1	4	3	18
$\frac{5}{8}$	$2\frac{7}{8}$	$2\frac{1}{16}$	$1\frac{21}{32}$	2	10	5	2	22
$\frac{11}{16}$	$3\frac{1}{4}$	$2\frac{1}{4}$	$1\frac{13}{16}$	3	1	6	4	27
$\frac{3}{4}$	$3\frac{5}{8}$	$2\frac{1}{2}$	2	3	12	7	7	32
$\frac{13}{16}$	$3\frac{3}{4}$	$2\frac{5}{8}$	$2\frac{1}{8}$	4	5	8	13	37
$\frac{7}{8}$	$4\frac{1}{8}$	$2\frac{15}{16}$	$2\frac{3}{8}$	4	18	10	0	43
$\frac{15}{16}$	$4\frac{3}{8}$	$3\frac{1}{8}$	$2\frac{1}{2}$	5	13	11	11	49
1	$4\frac{1}{2}$	$3\frac{1}{4}$	$2\frac{5}{8}$	6	11	13	8	56
$1\frac{1}{16}$	$4\frac{11}{16}$	$3\frac{1}{2}$	$2\frac{13}{16}$	7	6	14	8	63
$1\frac{1}{8}$	$5\frac{3}{16}$	$3\frac{11}{16}$	$2\frac{15}{16}$	8	4	16	14	71
$1\frac{3}{16}$	$5\frac{11}{16}$	$3\frac{7}{8}$	$3\frac{1}{8}$	9	2	18	11	79
$1\frac{1}{4}$	$5\frac{7}{8}$	4	$3\frac{3}{16}$	10	0	20	8	87
$1\frac{5}{16}$	$6\frac{1}{8}$	$4\frac{1}{2}$	$3\frac{5}{8}$	11	2	22	13	96
$1\frac{3}{8}$	$6\frac{7}{8}$	$4\frac{1}{2}$	$3\frac{5}{8}$	12	4	24	18	106
$1\frac{1}{2}$	$7\frac{1}{4}$	$4\frac{7}{8}$	$3\frac{7}{8}$	14	8	29	8	126

TABLE 24.—WORKING LOADS FOR CHAINS.

Single Sling Chains. *Double Sling Chains.*

Size of chain	Test load Admiralty strain.			Maximum working load.			Test load Admiralty strain (on the double).			Maximum working load.			Size of chain
ins.	tons	cwt.	qr.	tons	cwt.	qr.	tons	cwt.	qr.	tons	cwt.	qr.	ins.
3/16	–	8	2	–	5	0	–	17	0	–	10	0	3/16
1/4	–	15	0	–	9	0	1	10	0	–	18	0	1/4
5/16	1	2	2	–	13	2	2	5	0	1	7	0	5/16
3/8	1	12	2	–	19	2	3	5	0	1	19	0	3/8
7/16	2	5	0	1	7	0	4	10	0	2	14	0	7/16
1/2	3	0	0	1	16	0	6	0	0	3	12	0	1/2
9/16	3	15	0	2	5	0	7	10	0	4	10	0	9/16
5/8	4	12	2	2	15	2	9	5	0	5	11	0	5/8
11/16	5	12	2	3	7	2	11	5	0	6	15	0	11/16
3/4	6	15	0	4	1	0	13	10	0	8	2	0	3/4
13/16	7	18	0	4	15	0	15	16	0	9	10	0	13/16
7/8	9	2	2	5	9	2	18	5	0	10	19	0	7/8
15/16	10	10	0	6	6	0	21	0	0	12	12	0	15/16
1	12	0	0	7	4	0	24	0	0	14	8	0	1
1 1/16	13	10	0	8	2	0	27	0	0	16	4	0	1 1/16
1 1/8	15	2	2	9	1	2	30	5	0	18	3	0	1 1/8
1 3/16	16	18	0	10	3	0	33	16	0	20	6	0	1 3/16
1 1/4	18	15	0	11	5	0	37	10	0	22	10	0	1 1/4
1 5/16	20	12	2	12	5	0	41	5	0	24	10	0	1 5/16
1 3/8	22	12	2	13	10	0	45	5	0	27	0	0	1 3/8
1 1/2	27	0	0	15	0	0	54	0	0	30	0	0	1 1/2
1 5/8	31	12	2	17	5	0	63	5	0	34	10	0	1 5/8
1 3/4	36	15	0	20	0	0	73	10	0	40	0	0	1 3/4
1 7/8	42	3	2	23	0	0	84	7	0	46	0	0	1 7/8
2	48	0	0	26	0	0	96	0	0	52	0	0	2

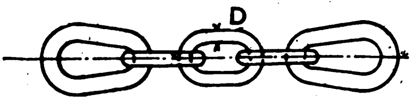


FIG. 36.

Special link to allow Shackle to pass through.

NOTE.—Particulars of special links at ends of chain are to be taken from table of double chain slings for corresponding size of chain.

TABLE 25.—SINGLE CHAIN SLINGS.

Safe load in tons	1/4	1/2	3/4	1	1 1/2	2	3	4	5	6	8	10
Dia. of Chain D	1/4	5/16	3/8	7/16	1/2	5/8	3/4	7/8	15/16	1	1 3/16	1 5/16

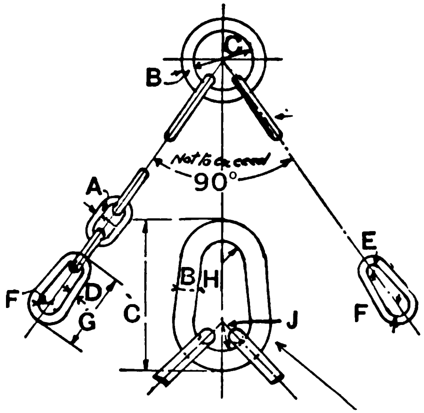


FIG. 37.

SPECIAL LINK TO REPLACE RING FOR LOADS FROM 6 TO 10 TONS.

TABLE 26.—DOUBLE CHAIN SLINGS.

Double Chain Slings and Rings.									
Safe load.	A	B		C		D	E	F	G
$\frac{1}{2}$ ton	$\frac{1}{4}$	$\frac{5}{8}$		2		$\frac{3}{8}$	$\frac{1}{4}$	$\frac{7}{16}$	$2\frac{3}{4}$
$\frac{3}{4}$,,	$\frac{5}{16}$	$\frac{3}{4}$		$2\frac{1}{2}$		$\frac{1}{2}$	$\frac{5}{16}$	$\frac{1}{2}$	$3\frac{1}{4}$
1 ,	$\frac{3}{8}$	$\frac{7}{8}$		$2\frac{3}{4}$		$\frac{9}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$3\frac{5}{8}$
$1\frac{1}{2}$ tons	$\frac{7}{16}$	1		3		$\frac{5}{8}$	$\frac{3}{8}$	$\frac{5}{8}$	4
2	$\frac{1}{2}$	$1\frac{1}{8}$		$3\frac{1}{2}$		$\frac{3}{4}$	$\frac{7}{16}$	$\frac{3}{4}$	$4\frac{3}{4}$
3 ,	$\frac{5}{8}$	$1\frac{1}{4}$		$3\frac{7}{8}$		$\frac{7}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$5\frac{1}{2}$
4 ,,	$\frac{11}{16}$	$1\frac{1}{2}$		$4\frac{3}{8}$		$1\frac{1}{16}$	$\frac{9}{16}$	1	6
5 ,,	$\frac{13}{16}$	$1\frac{5}{8}$		$4\frac{3}{4}$		$1\frac{1}{8}$	$\frac{5}{8}$	$1\frac{1}{8}$	$6\frac{1}{2}$
		B*	C*	H	J				
6 ,,	$\frac{7}{8}$	$1\frac{5}{8}$	$9\frac{3}{4}$	$1\frac{1}{2}$	2	$1\frac{5}{16}$	$\frac{3}{4}$	$1\frac{5}{8}$	$7\frac{1}{4}$
8 ,,	1	$1\frac{7}{8}$	$11\frac{1}{4}$	$1\frac{5}{8}$	$2\frac{1}{4}$	$1\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{2}$	$8\frac{1}{4}$
10 ,,	$1\frac{1}{8}$	$2\frac{1}{8}$	$12\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{2}$	$1\frac{5}{8}$	1	$1\frac{5}{8}$	9

NOTE.—The above table is for double sling chains. When using treble sling chains take the proportions as for two-thirds load required.

NOTE.—The angle between chains must not exceed 90° for the loads given in this table.

TABLE 27.—STUD-LINKED CHAIN CABLE.

Size of chain.	Number of links in each length of cable.	Wt. of 100 fathoms of cable with fittings.	Weight of components				Admiralty proof load.
			Shackle	End link.	Intermediate link.	Ordinary link.	
in.		cwt.	lb.	lb.	lb.	lb.	tons.
⅜	595	7	·44	·22	·21	·18	2 17/32
7/16	509	10	·7	·34	·33	·28	3½
½	445	13	1·04	·53	·47	·41	4½
9/16	395	16	1·49	·75	·70	·58	5½
⅝	355	20	2·04	1·03	1·03	·8	7
11/16	321	25	2·72	1·37	1·29	1·1	8½
¾	295	29	3·53	1·79	1·68	1·4	10⅛
⅞	253	40	5·61	2·84	2·66	2 ·2	13¾
15/16	237	46	6·89	3·5	3·29	2·63	15⅞
1	221	53	8·37	4·25	4	3·2	18
1⅛	195	64	11·87	6·10	5·7	4·5	22¾
1¼	175	75	16·31	8·32	7·75	6·1	28⅛
1⅜	157	91	21·75	11	10·37	8·2	34
1½	145	108	28	14·34	13·5	10·5	40½
1⅝	133	127	36	18·25	17·2	13·4	47½
1¾	123	147	44·9	22·78	21·5	16·75	55⅛
1⅞	115	169	55·25	28	26·33	20·6	63¼
2	107	192	67	34	32	25	72
2⅛	99	217	80	40·75	38·3	30	81¼
2¼	95	243	95	48·4	45·5	35·6	91⅛
2⅜	89	271	112	56·9	53·5	41·9	101½
2½	85	300	130	66·4	62·5	48·8	112½
2 9/16	83	315	140	71·5	66·3	52·6	116¾
2 11/16	77	346	162·4	82·5	77·4	60·7	125⅛
2¾	75	363	174	88·38	83	65	129⅛
2⅞	73	397	199	101·1	95	74·3	137⅛
3	69	432	226·1	114·75	108	84·38	145¾
3¼	63	507	287·5	145·9	137	107·25	161⅝
3½	59	588	359	182·25	171·5	134	176⅜
3¾	55	675	441	224	210·5	164·75	189¾
4	51	768	535·7	272	256	204·8	201⅛

SHAFT COUPLINGS.

TABLE 27.—SOLID FLANGE COUPLINGS.

A inches.	B inches.	C inches.	D inches.	E inches.	Number of bolts.
$1\frac{1}{4}$	$2\frac{7}{8}$	$\frac{1}{2}$	$2\frac{1}{8}$	$\frac{3}{8}$	4
$1\frac{1}{2}$	$3\frac{1}{2}$	$\frac{5}{8}$	$2\frac{1}{2}$	$\frac{7}{16}$	4
2	$4\frac{3}{4}$	$\frac{7}{8}$	$3\frac{1}{2}$	$\frac{5}{8}$	5
$2\frac{1}{2}$	$5\frac{7}{8}$	$1\frac{1}{8}$	$4\frac{1}{4}$	$\frac{7}{8}$	5
$2\frac{3}{4}$	$6\frac{1}{2}$	$1\frac{1}{4}$	$4\frac{5}{8}$	1	5
3	$7\frac{1}{8}$	$1\frac{3}{8}$	$5\frac{1}{8}$	1	6
$3\frac{1}{2}$	$8\frac{1}{4}$	$1\frac{1}{2}$	$5\frac{3}{4}$	$1\frac{1}{8}$	6
4	$9\frac{1}{2}$	$1\frac{5}{8}$	6	$1\frac{1}{4}$	6
$4\frac{1}{4}$	10	$1\frac{3}{4}$	$6\frac{7}{8}$	$1\frac{1}{4}$	6
$4\frac{3}{4}$	$11\frac{1}{8}$	2	$7\frac{7}{8}$	$1\frac{1}{2}$	6
5	$11\frac{3}{4}$	2	$8\frac{1}{2}$	$1\frac{1}{2}$	6
$5\frac{1}{2}$	13	$2\frac{1}{4}$	$9\frac{3}{8}$	$1\frac{3}{4}$	6
6	$14\frac{1}{8}$	$2\frac{1}{2}$	$10\frac{1}{4}$	$1\frac{7}{8}$	6

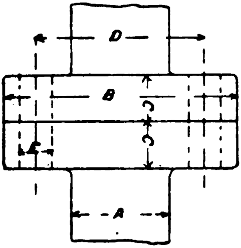


FIG. 38.

TABLE 28.—CAST-IRON FLANGE COUPLINGS.

A ins.	B ins.	C ins.	D ins.	E ins.	F ins.	G ins.	H ins.	J ins.	K ins.	L ins.	M ins.	N ins.	O ins.	Number of bolts.
1¼	8	6¼	2⅜	1⅝	1¼	2½	3/16	3	4¾	4¾	¾	⅜	⅝	4
1½	8⅝	7	3⅛	1¾	1¼	3¼	3/16	3½	5½	5½	⅞	⅜	⅝	4
2	9½	8¼	3½	2	1½	3½	¼	4	6¼	6	1	½	¾	4
2½	10⅝	9	4	2¼	1½	4	¼	4¾	7	6⅝	1⅛	½	¾	4
2¾	11¾	10¼	4⅜	2⅜	1¾	4⅜	5/16	5½	7⅞	7½	1¼	⅝	⅞	4
3	13¼	11½	5	2⅝	1¾	4¾	5/16	6¼	9	8⅝	1½	⅝	⅞	5
3½	15	12½	5½	2⅞	2	5½	⅜	7	10¼	9⅞	1⅝	⅝	1	5
4	16½	13½	6	3⅛	2¼	6¼	⅜	7½	11½	10⅝	1¾	11/16	1	6
4¼	17¾	14⅛	6½	3¼	2⅜	6⅝	7/16	8¼	12½	11½	1⅞	11/16	1⅛	6
4¾	19¼	15¾	7	3½	2½	7½	7/16	9	13¾	12½	2	11/16	1¼	6
5	20¾	17¼	7½	3¾	2¾	7¾	½	9½	15	13¾	2⅛	13/16	1⅜	6
5½	22	18¾	8¼	4	3	8⅝	½	10¼	16	15	2 3/16	13/16	1½	6
6	23½	20	9	4¼	3¼	9	9/16	11	17¼	16	2¼	13/16	1½	6

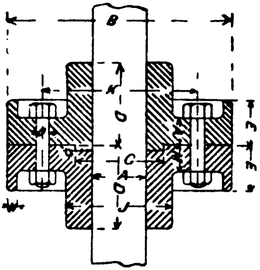


FIG. 39.

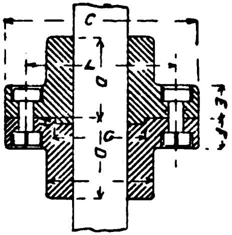


FIG. 40.

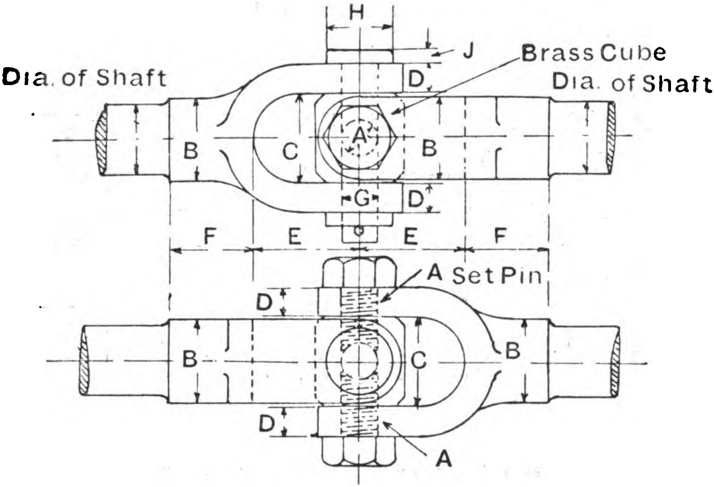


FIG. 41.—STEEL UNIVERSAL JOINTS.

TABLE 29.—PROPORTIONS OF STEEL UNIVERSAL JOINTS.

Set pins A to be screwed in before hole for bolt G is drilled. Bolt G will then prevent set pins from running back.

Dia. of Shaft.	A	B	C	D	E	F	G	H	J
ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.
⅞	½	1⅛	1¼	7/16	1½	1¼	½	⅞	3/16
1⅛	⅝	1⅜	1½	9/16	1⅞	1½	⅝	1⅛	3/16
1½	¾	1¾	1⅞	⅝	2¼	1¾	¾	1⅜	¼
1¾	⅞	2	2⅛	11/16	2½	2	⅞	1 9/16	¼
2	1	2¼	2⅜	¾	2¾	2¼	1	1¾	5/16
2½	1¼	2¾	2⅞	⅞	3¼	2⅝	1¼	2⅛	⅜
3	1½	3¼	3⅜	1⅛	3¾	3	1½	2½	½
3½	1¾	3¾	3⅞	1⅜	4¾	3¼	1¾	3	⅝
4	2	4¼	4⅜	1⅝	5¼	3½	2	3¼	⅝
4½	2¼	4¾	4⅞	2	5¾	4	2¼	3½	¾
5	2½	5¼	5⅜	2¼	6¼	4½	2½	3⅞	¾

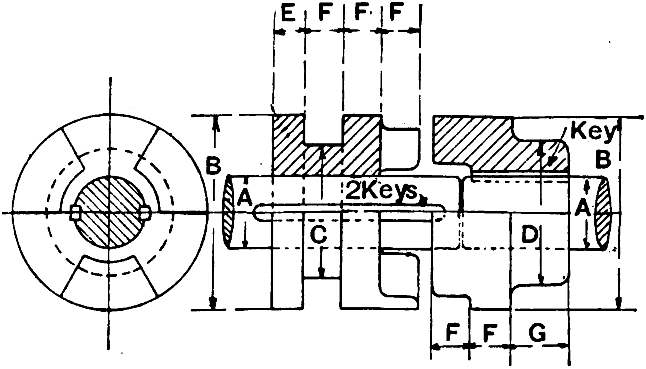


FIG. 42.—CLAW COUPLINGS.

TABLE 30.—CLAW COUPLING PROPORTIONS.

A	B	C	D	E	F	G
ins.	ins.	ins.	ins.	ins.	ins.	ins.
$1\frac{1}{2}$	5	$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{5}{8}$	$\frac{7}{8}$	$1\frac{1}{2}$
$1\frac{3}{4}$	$5\frac{1}{2}$	$3\frac{7}{8}$	4	$\frac{3}{4}$	1	$1\frac{3}{4}$
2	6	$4\frac{1}{8}$	$4\frac{3}{8}$	$\frac{13}{16}$	$1\frac{1}{8}$	$1\frac{7}{8}$
$2\frac{1}{4}$	$6\frac{1}{2}$	$4\frac{5}{8}$	$4\frac{1}{2}$	$\frac{7}{8}$	$1\frac{1}{4}$	2
$2\frac{1}{2}$	7	$4\frac{7}{8}$	$5\frac{1}{4}$	1	$1\frac{3}{8}$	$2\frac{1}{4}$
$2\frac{3}{4}$	$7\frac{1}{2}$	$5\frac{1}{4}$	$5\frac{3}{4}$	1	$1\frac{1}{2}$	$2\frac{3}{8}$
3	8	$5\frac{1}{2}$	$6\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{5}{8}$	$2\frac{1}{2}$
$3\frac{1}{4}$	$8\frac{1}{2}$	6	$6\frac{5}{8}$	$1\frac{1}{8}$	$1\frac{3}{4}$	$2\frac{5}{8}$
$3\frac{1}{2}$	9	$6\frac{3}{8}$	7	$1\frac{1}{8}$	$1\frac{7}{8}$	$2\frac{3}{4}$
$3\frac{3}{4}$	$9\frac{1}{2}$	$6\frac{3}{4}$	$7\frac{1}{2}$	$1\frac{1}{4}$	2	$2\frac{7}{8}$
4	10	7	$7\frac{7}{8}$	$1\frac{1}{4}$	$2\frac{1}{8}$	3
$4\frac{1}{4}$	$10\frac{1}{2}$	$7\frac{5}{8}$	$8\frac{1}{4}$	$1\frac{3}{8}$	$2\frac{1}{4}$	$3\frac{1}{4}$
$4\frac{1}{2}$	11	$7\frac{3}{4}$	$8\frac{5}{8}$	$1\frac{1}{2}$	$2\frac{3}{8}$	$3\frac{3}{8}$
$4\frac{3}{4}$	$11\frac{1}{2}$	8	$9\frac{1}{8}$	$1\frac{3}{4}$	$2\frac{1}{2}$	$3\frac{1}{2}$
5	12	$8\frac{1}{8}$	$9\frac{1}{2}$	$1\frac{7}{8}$	$2\frac{5}{8}$	$3\frac{5}{8}$
$5\frac{1}{4}$	$12\frac{1}{2}$	$8\frac{1}{4}$	10	2	$2\frac{3}{4}$	$3\frac{7}{8}$
$5\frac{1}{2}$	13	$8\frac{5}{8}$	$10\frac{3}{8}$	$2\frac{1}{8}$	$2\frac{7}{8}$	4
$5\frac{3}{4}$	$13\frac{1}{2}$	$8\frac{3}{4}$	$10\frac{3}{4}$	$2\frac{1}{4}$	3	$4\frac{1}{8}$
6	14	9	$11\frac{1}{8}$	$2\frac{3}{8}$	$3\frac{1}{4}$	$4\frac{1}{4}$

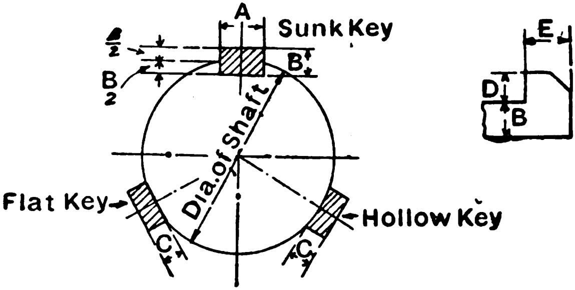


FIG. 43.—SHAFT KEYS.

TABLE 31.—PROPORTIONS OF KEYS FOR SHAFTS

Dia. of Shaft.	A	B	C	D	E
ins.	ins.	ins.	ins.	ins.	ins.
$\frac{3}{4}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{5}{32}$	$\frac{1}{4}$
1	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{9}{32}$
$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{11}{32}$
$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{3}{8}$
$1\frac{3}{4}$	$\frac{9}{16}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{9}{32}$	$\frac{7}{16}$
2	$\frac{5}{8}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{15}{32}$
$2\frac{1}{4}$	$\frac{11}{16}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{11}{32}$	$\frac{1}{2}$
$2\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{9}{16}$
$2\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{16}$	$\frac{5}{16}$	$\frac{13}{32}$	$\frac{5}{8}$
3	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{5}{16}$	$\frac{7}{16}$	$\frac{21}{32}$
$3\frac{1}{2}$	1	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$
4	$1\frac{1}{8}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{27}{32}$
$4\frac{1}{2}$	$1\frac{1}{4}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{15}{16}$
5	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{11}{16}$	$1\frac{1}{32}$
6	$1\frac{5}{8}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{13}{16}$	$1\frac{1}{4}$
7	$1\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{15}{16}$	$1\frac{3}{8}$
8	$2\frac{1}{8}$	$\frac{15}{16}$	$\frac{3}{4}$	$1\frac{1}{16}$	$1\frac{5}{8}$
9	$2\frac{3}{8}$	1	$\frac{7}{8}$	$1\frac{3}{16}$	$1\frac{3}{4}$

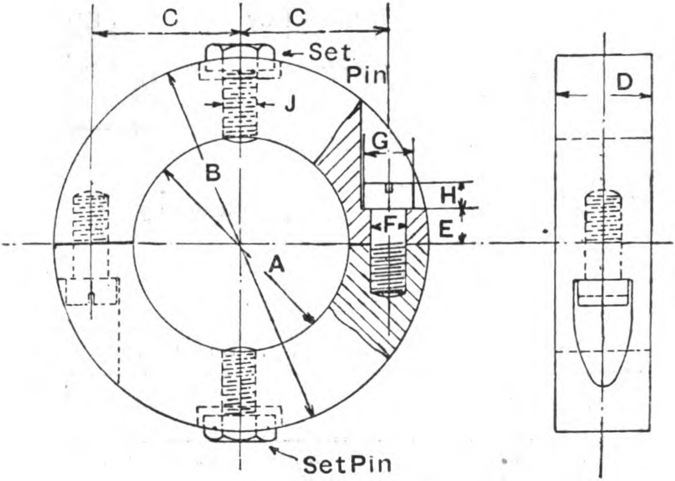


Fig. 44.—Shaft Collars.

Table 32.—Cast Iron Split Collars for Shafts.

A	B	C	D	E	F	G	H	Set Pins.	
								No.	J
ins.	ins.	ins.	ins.	ins.	ins.	ins.	ins.		ins.
1	$2\frac{1}{4}$	$\frac{13}{16}$	$\frac{3}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{7}{16}$	$\frac{1}{4}$	1	$\frac{3}{8}$
$1\frac{1}{2}$	$3\frac{1}{8}$	$1\frac{1}{8}$	1	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{9}{16}$	$\frac{5}{16}$	1	$\frac{3}{8}$
2	$3\frac{3}{4}$	$1\frac{7}{16}$	1	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{9}{16}$	$\frac{5}{16}$	1	$\frac{1}{2}$
$2\frac{1}{2}$	5	$1\frac{7}{8}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{8}$	1	$\frac{1}{2}$
3	$5\frac{3}{8}$	$2\frac{1}{8}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{8}$	1	$\frac{5}{8}$
$3\frac{1}{2}$	$6\frac{3}{8}$	$2\frac{1}{2}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{15}{16}$	$\frac{7}{16}$	1	$\frac{5}{8}$
4	$6\frac{7}{8}$	$2\frac{3}{4}$	$1\frac{3}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{15}{16}$	$\frac{7}{16}$	2	$\frac{5}{8}$
$4\frac{1}{2}$	$7\frac{7}{8}$	$3\frac{1}{8}$	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{16}$	$\frac{9}{16}$	2	$\frac{3}{4}$
5	9	$3\frac{1}{2}$	$1\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{11}{16}$	2	$\frac{3}{4}$
$5\frac{1}{2}$	$9\frac{1}{2}$	$3\frac{3}{4}$	$1\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{11}{16}$	2	$\frac{3}{4}$
6	$10\frac{1}{2}$	$4\frac{1}{8}$	$1\frac{7}{8}$	1	1	$1\frac{7}{16}$	$\frac{3}{4}$	2	$\frac{3}{4}$
$6\frac{1}{2}$	11	$4\frac{3}{8}$	$1\frac{7}{8}$	1	1	$1\frac{7}{16}$	$\frac{3}{4}$	2	$\frac{7}{8}$
7	$11\frac{1}{2}$	$4\frac{5}{8}$	$1\frac{7}{8}$	1	1	$1\frac{7}{16}$	$\frac{3}{4}$	2	$\frac{7}{8}$
8	13	$5\frac{1}{4}$	2	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{5}{8}$	$\frac{13}{16}$	2	$\frac{7}{8}$

SECTION V

WATER AND AIR DATA

WATER AND AIR DATA.

HYDRAULIC MEMORANDA

1 Imperial or English gallon of Water ... ... = 277·464 cubic inches;
= 0·160 cubic foot;
= 10·000 pounds (lbs.);
= 4·547 litres.

1 United States gallon of Water ... ... = 231·000 cubic inches;
= 0·133 cubic foot;
= 8·330 lbs.;
= 0·830 English gallon;
= 3·8 litres.

1 cubic inch of Water ... = 0·03607 lb.
= 0·003607 English gallon;
= 0·004329 United States gallon.

1 cubic foot of Water ... = 6·235 English gallons;
= 7·48 United States gallons,
= 7·901 New York gallons;
= 28·375 litres;
= 0·0283 cubic metre;
= 62·35 lbs.;
= 0·557 cwt.;
= 0·028 ton.

1 lb. of Water ... ... = 27·464 cubic inches;
= 0·10 English gallon;
= 0·12 United States gallon;
= 0·4537 kilo.

1 cwt. of Water ... ... = 11·2 English gallons;
= 13·44 United States gallons;
= 1·8 cubic feet.

1 ton of Water ... ... = 35·9 cubic feet;
= 224·0 English gallons;
= 268·8 United States gallons;
= 1000 litres (approximately);
= 1 cubic metre (approximately).

1 litre of Water ... ... = 0·22 English gallon;
= 0·264 United States gallon;
= 0·275 New York gallon;
= 61 cubic inches;
= 0·0353 cubic foot.

1 cubic metre or kilometre of Water ... ... = 220 English gallons;
= 264 United States gallons;
= 1·308 cubic yards;
= 35·31 cubic feet;
= 61,028 cubic inches;
= 1000 kilos.;
= 1 ton (approximately);
= 1000 litres.
1 kilo. of Water ... ... = 2·204 lbs.
1 miner's inch ... ... = The quantity of water which will pass in 24 hours through an aperture of one square inch under a head of 6 inches;
= 2160 cubic feet, or 12,960 gallons of water, or about 58 tons.

To find pressure of water in lbs. per square inch, when the head in feet is given.

$P = H \times 0{\cdot}433$; or nearly $0{\cdot}44$;
When P = pressure in lbs. per square inch;
H = head of water in feet.

To find head of water in feet, when pressure in lbs. per square inch is given.

$H = P \times 2{\cdot}307$; usually taken as $2{\cdot}3$.

To find the pressure in lbs. per square foot when the head in feet is given.

$P_1 = H \times 62{\cdot}4$;
When P_1 = pressure in lbs. per square foot.

To find the discharge of water over a rectangular weir.

$$C = 5{\cdot}15\sqrt{d^3}.$$

When C = cubic feet of water discharged per minute for every foot in width of weir;
d = depth from surface of water where at rest, to the top of sill in inches.

$$H = \frac{d}{12}.$$

H = depth in feet.

$C = 214\sqrt{H^3 + 0{\cdot}035 \times V^2 \times H}$; if the water approaches the sill with a velocity $= V =$ velocity in feet per second.

TABLE 1.—QUANTITY OF WATER IN CUBIC FEET PER MINUTE DELIVERED FOR EACH INCH OF WIDTH OF RECTANGULAR WEIR.

Inch. depth of Weir.	Fraction of an Inch.							
	0	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$
1	0·40	0·47	0·56	0·65	0·74	0·83	0·92	1·03
2	1·14	1·25	1·36	1·47	1·59	1·71	1·84	1·96
3	2·09	2·22	2·36	2·60	2·64	2·78	2·93	3·06
4	3·22	3·38	3·53	3·69	3·85	4·01	4·17	4·34
5	4·51	4·68	4·85	5·02	5·20	5·38	5·56	5·74
6	5·92	6·10	6·30	6·49	6·68	6·87	7·07	7·27
7	7·46	7·67	7·87	8·07	8·28	8·49	8·70	8·91
8	9·12	9·33	9·55	9·77	9·99	10·21	10·43	10·66
9	10·88	11·11	11·34	11·57	11·80	12·04	12·27	12·51
10	12·75	13·15	13·23	13·47	13·72	13·96	14·21	14·46
11	14·71	14·96	15·21	15·46	15·72	15·98	16·24	16·49
12	16·76	17·02	17·28	17·55	17·82	18·08	18·35	18·62
13	18·89	19·17	19·44	19·72	20·00	20·27	20·56	20·83
14	21·22	21·40	21·68	21·97	22·26	22·55	22·84	23·13
15	23·42	23·71	24·01	24·30	24·60	24·90	25·19	25·50
16	25·80	26·10	26·41	26·71	27·02	27·32	27·63	27·94
17	28·56	28·57	28·88	29·19	29·51	29·51	30·14	30·45
18	30·78	31·11	31·43	31·75	32·07	32·40	32·72	33·50

TABLE 2.—QUANTITY OF WATER IN GALLONS PER MINUTE DELIVERED OVER A V-NOTCH.

Head in inch.	Decimal parts of an Inch.									
	·0	·1	·2	·3	·4	·5	·6	·7	·8	·9
1	1·90	2·42	3·00	3·67	4·41	5·24	6·16	7·17	8·27	9·47
2	10·76	12·16	13·66	15·27	16·98	18·81	20·75	22·80	24·97	27·26
3	29·67	32·20	34·86	37·65	40·57	43·62	46·80	50·12	53·57	57·17
4	60·90	64·78	68·80	72·97	77·29	81·75	86·37	91·14	96·08	101·15

To find the velocity of the discharge of water in feet per second flowing through a sluice or penstock.

$$V = 8{\cdot}025\ \sqrt{H};$$

$$H = \frac{\sqrt{V}}{64{\cdot}4};$$

$$C = A \times T \times 5\ \sqrt{H};$$

$$G = A \times T \times 31{\cdot}5\ \sqrt{H};$$

When V = velocity of discharge in feet per second ;
H = height from the centre of the orifice to the surface of the water in feet ;
C = number of cubic feet discharged per second ;
G = number of gallons discharged per second ;
A = area of orifice in square feet ;
T = time in seconds.

To find theoretical horse-power of water.

$$THP = C \times H \times 0{\cdot}001892;$$

When THP = theoretical horse-power ;
C = cubic feet of water obtainable per minute ;
H = head of water from the tail-race in feet.

To find theoretical velocity of water due to given heads.

$$V = 8{\cdot}025\ \sqrt{H};$$

$$V_1 = 482\ \sqrt{H};$$

When V = theoretical velocity in feet per second ;
H = head of water in feet ;
V_1 = theoretical velocity in feet per minute ;

To find the flow of water through boreholes

(Greenwell's Rule).

$$V = 2837{\cdot}5 \times \sqrt{\frac{P \times D}{L}};$$

When V = velocity at point of issue in feet per minute ;
P = pressure in feet ;
D = diameter of borehole in feet ;
L = length of borehole in feet ;

and

$$G = 6{\cdot}25\ V \times 0{\cdot}7854 \times D^2;$$

When G = gallons per minute.

or

(Eylelwein's Rule).

$$W = 4{\cdot}71 \sqrt{\frac{D^5 \times H}{L}}$$

When W = cubic feet of water discharged per minute ;
D = diameter of borehole in inches ;
H = head of water in feet ;
L = length of borehole in feet.

Flow of water through 2½-inch hose

(E. B. Weston's Rule).

1. For hose between 90 and 100 feet long—

$$V = \sqrt{\frac{2\,g\,H}{1 - 0{\cdot}0256\ D^4 + \left(0{\cdot}0087 + \frac{{\cdot}0505}{\sqrt{V_1}}\right) + {\cdot}12288\ D^4\ L}}.$$

2. For all lengths of hose (a reliable rule) :—

$$V = \sqrt{\frac{H}{{\cdot}0155463 - {\cdot}000398D^4 + {\cdot}0000362962D^4L}};$$

When V = velocity of efflux in feet per second ;
H = head in feet indicated by gauge ;
D = diameter of coupling in inches ;
V_1 = velocity in 2½ inch hose ;
F = loss of head in feet.

$F = 0{\cdot}001135\ V_{11}^2$, in 45 experiments with ring nozzles.
$F = 0{\cdot}0009639\ V_{11}^2$, in 35 experiments with smooth nozzles.

1 Imperial or English gallon of water... ... ... = ·36 Spanish Arroba ;
= 2·707 Russian Vedros ;
= ·8729 Turkish Almud.
1 Spanish Arroba ... ... = 2·77 English gallons.
1 Russian Vedro ... ... = ·37 ,, ,,
1 Turkish Almud ... ... = 1·145 ,, ,,
40 Russian Vedros ... ... = 1 Russian Volchka.

TABLE 3.—SPOUTING VELOCITY AND DISCHARGE OF WATER THROUGH RECTANGULAR OPENINGS.

Head of water in inches.	Velocity per second in inches and decimals.	Cubic feet discharged per minute. Area of opening 1 inch.	Head of water in inches.	Velocity per second in inches and decimals.	Cubic feet discharged per minute. Area of opening 1 inch.	Head of water in inches.	Velocity per second in inches and decimals.	Cubic feet discharged per minute. Area of opening 1 inch.
1	17·64	0·61	15	68·33	2·37	29	95·00	3·35
2	24·95	1·06	16	70·57	2·45	30	96·63	3·41
3	30·55	1·16	17	72·74	2·53	31	98·22	3·46
4	35·28	1·22	18	74·85	2·60	32	99·80	3·52
5	39·45	1·37	19	76·90	2·67	33	101·34	3·57
6	43·21	1·50	20	78·90	2·75	34	102·87	3·63
7	46·68	1·62	21	80·84	2·87	35	104·37	3·67
8	49·60	1·73	22	82·75	2·93	36	105·85	3·72
9	52·92	1·84	23	84·61	3·00	37	107·31	3·77
10	55·79	1·94	24	86·43	3·06	38	108·75	3·82
11	58·51	2·03	25	88·21	3·12	39	110·17	3·87
12	61·11	2·12	26	89·96	3·18	40	111·58	3·93
13	63·61	2·21	27	91·67	3·24			
14	66·01	2·29	28	93·35	3·30			

To find the discharge of water over a V-notch
(Dr. Thompson's Formula).

$$Q = 0{\cdot}317h^{\frac{5}{2}}.$$

When Q = cubic feet of water per minute ;
h = head in inches, measured vertically from still water-level of the pool to the vertex of the notch.

The sides of the notch to have a slope of 45 degrees with the horizon.

When a = ratio of half breadth of notch at surface water-level to h, then $2ah$ = breadth of notch at this level.

If $a = 1$, then $Q = 2{\cdot}54h^{\frac{5}{2}}$ = cubic feet per second.
$Q = 152{\cdot}4h^{\frac{5}{2}}$ = ,, ,, minute.
If a = 2, then $Q = 5{\cdot}3h^{\frac{5}{2}}$ = ,, ,, second.
$Q = 31Sh^{\frac{5}{2}}$ = ,, ,, minute.

TABLE 4.—THEORETICAL VELOCITY OF WATER DUE TO GIVEN HEADS.

Head of water in feet.	Theoretical velocity in feet per second.	Theoretical velocity in feet per minute.	Head of water in feet.	Theoretical velocity in feet per second.	Theoretical velocity in feet per minute.	Head of water in feet.	Theoretical velocity in feet per second.	Theoretical velocity in feet per minute.
1	8·025	481·5	26	40·919	2455·1	51	57·309	3438·5
2	11·345	681·7	27	41·699	2501·9	52	57·869	3472·1
3	13·899	833·9	28	42·464	2547·8	53	58·422	3505·3
4	16·050	963·0	29	43·215	2592·9	54	58·971	3538·2
5	17·944	1076·6	30	43·954	2637·2	55	59·515	3570·9
6	19·657	1179·4	31	44·681	2680·8	56	60·053	3603·2
7	21·232	1273·6	32	45·396	2723·7	57	60·587	3635·2
8	22·698	1361·8	33	46·100	2766·0	58	61·116	3666·9
9	24·075	1444·5	34	46·793	2783·0	59	61·641	3698·4
10	25·377	1522·6	35	47·476	2848·5	60	62·161	3729·6
11	26·615	1596·9	36	48·150	2889·0	61	62·677	3760·6
12	27·799	1667·9	37	48·814	2928·8	62	63·188	3791·3
13	28·934	1736·0	38	49·469	2968·1	63	63·696	3821·7
14	30·026	1801·6	39	50·116	3006·9	64	64·200	3852·0
15	31·080	1864·8	40	50·754	3045·2	65	64·699	3881·9
16	32·100	1926·0	41	51·385	3083·1	66	65·195	3911·7
17	33·087	1985·2	42	52·007	3120·4	67	65·687	3941·2
18	34·047	2042·8	43	52·623	3157·4	68	66·175	3970·3
19	34·980	2098·8	44	53·231	3193·9	69	66·660	3999·6
20	35·888	2153·3	45	53·833	3229·9	70	67·141	4028·5
21	36·775	2206·5	46	54·427	3265·6	71	67·619	4057·1
22	37·640	2258·4	47	55·016	3301·0	72	68·094	4085·6
23	38·486	2309·1	48	55·598	3335·8	73	68·565	4113·9
24	39·314	2358·8	49	56·175	3370·5	74	69·033	4142·0
25	40·125	2407·5	50	56·745	3404·7	75	69·498	4169·9

TABLE 4.—THEORETICAL VELOCITY OF WATER DUE TO GIVEN HEADS (*Continued*).

Head of water in feet.	Theoretical velocity in feet per second.	Theoretical velocity in feet per minute.	Head of water in feet.	Theoretical velocity in feet per second.	Theoretical velocity in feet per minute.	Head of water in feet.	Theoretical velocity in feet per second.	Theoretical velocity in feet per minute.
76	69·960	4197·6	105	82·231	4933·9	230	121·70	7302·3
77	70·419	4225·1	110	84·166	5050·0	235	123·02	7381·2
78	70·874	4252·4	115	86·058	5163·5	240	124·32	7459·3
79	71·327	4279·6	120	87·909	5274·5	245	125·60	7536·6
80	71·777	4306·6	125	89·722	5383·3	250	126·88	7613·1
81	72·225	4333·5	130	91·499	5489·9	255	128·15	7648·8
82	72·673	4360 4	135	93·242	5594·5	260	129·39	7763 9
83	73·111	4386·6	140	94·953	5697·2	265	130·63	7837·6
84	73·550	4413·0	145	96·633	5798·0	270	131·86	7911·8
85	73·986	4439·2	150	98·285	5897·1	275	133·08	7984·8
86	74·420	4465·2	155	99·909	5994·5	230	134·28	°057·0
87	74·852	4491·1	160	101·50	6090·5	285	135·48	8128·6
88	75·281	4516·8	165	103·08	6184·9	290	136·66	8199·6
89	75·707	4542·4	170	104·63	6277·9	295	137·83	8270·1
90	76·131	4567·9	175	106·16	6369·6	300	138·99	8339·8
91	76·553	4593·2	180	107·66	6460·0	305	140·15	8409·0
92	76·973	4618·3	185	109·15	6549·1	310	141·29	8477·6
93	77·390	4643·4	190	110·61	6637·0	315	142·42	8545·6
94	77·805	4668·3	195	112·06	6723·7	320	143·55	8613·3
95	78·217	4693·0	200	113·49	6809·4	325	144·67	8690·4
96	78·628	4717·7	205	114·90	6894·0	330	145·78	8760·9
97	79·037	4742·2	210	116·29	6977·6	335	146·88	8812·9
98	79·443	4766·6	215	117·66	7060·1	340	147·97	8878·4
99	79·847	4790·8	220	119·03	7141·8	345	149·06	8943·5
100	80·250	4815·0	225	120·00	7222·5	350	150·13	9007·9

TABLE 5.—PRESSURE OF WATER AT DIFFERENT HEADS IN POUNDS PER SQUARE INCH.

Head of water in feet.	Head of water in yards.	Head of water in fathoms.	Head of water in metres.	Pressure of water in lbs. per square inch.	Head of water in feet.	Head of water in yards.	Head of water in fathoms.	Head of water in metres.	Pressure of water in lbs. per square inch.
5	1·66	0·83	1·52	2·16	200	66·6	33·3	60·9	86·6
10	3·33	1·66	3·04	4·33	205	68·3	34·1	62·4	88·8
15	5·00	2·50	4·57	6·49	210	70·0	35·0	64·0	90·9
20	6·66	3·33	6·09	8·66	215	71·6	35·8	65·5	93·1
25	8·33	4·16	7·62	10·80	220	73·3	36·6	67·0	95·3
30	10·00	5·00	9·14	12·90	225	75·0	37·5	68·5	97·4
35	11·60	5·83	10·60	15·10	230	76·6	38·3	70·1	99·6
40	13·30	6·66	12·1	17·3	235	78·3	39·1	71·6	101·8
45	15·00	7·50	13·7	19·4	240	80·0	40·0	73·1	103·9
50	16·60	8·33	15·2	21·6	245	81·6	40·8	74·6	106·1
55	18·30	9·16	16·7	23·8	250	83·3	41·6	76·2	108·3
60	20·00	10·0	18·2	25·9	255	85·0	42·5	77·7	110·4
65	21·6	10·8	19·8	28·1	260	86·6	43·3	79·2	112·6
70	23·3	11·6	21·3	30·3	265	88·3	44·1	80·7	114·8
75	25·0	12·5	22·8	32·4	270	90·0	45·0	82·2	116·9
80	26·6	13·3	24·3	34·6	275	91·6	45·8	83·8	119·1
85	28·3	14·1	25·9	36·8	280	93·3	46·6	85·3	121·3
90	30·0	15·0	27·4	38·9	285	95·0	47·5	86·8	123·4
95	31·6	15·8	28·9	41·1	290	96·6	48·3	88·3	125·6
100	33·3	16·6	30·4	43·3	295	98·3	49·1	89·9	127·8
105	35·0	17·5	32·0	45·4	300	100·0	50·0	91·4	129·9
110	36·6	18·3	33·5	47·6	305	101·6	50·8	92·9	132·1
115	38·3	19·1	35·0	49·8	310	103·3	51·6	94·4	134·3
120	40·0	20·0	36·5	51·9	315	105·0	52·5	96·0	136·4
125	41·6	20·8	38·1	54·1	320	106·6	53·3	97·5	138·6
130	43·3	21·6	39·6	56·3	325	108·3	54·1	99·0	140·8
135	45·0	22·5	41·1	58·4	330	110·0	55·0	100·5	142·9
140	46·6	23·3	42·6	60·6	335	111·6	55·8	102·1	145·1
145	48·3	24·1	44·1	62·8	340	113·3	56·6	103·6	147·3
150	50·0	25·0	45·7	64·9	345	115·0	57·5	105·1	149·4
155	51·6	25·8	47·2	67·1	350	116·6	58·3	106·6	151·6
160	53·3	26·6	48·7	69·3	355	118·3	59·1	108·2	153·8
165	55·0	27·5	50·2	71·4	360	120·0	60·0	109·7	155·9
170	56·6	28·3	51·8	73·6	365	121·6	60·8	111·2	158·1
175	58·3	29·1	53·3	75·8	370	123·3	61·6	112·7	160·3
180	60·0	30·0	54·8	77·9	375	125·0	62·5	114·3	162·4
185	61·6	30·8	56·3	80·1	380	126·6	63·3	115·8	164·6
190	63·3	31·6	57·9	82·3	385	128·3	64·1	117·3	166·8
195	65·0	32·5	59·4	84·4	390	130·0	65·0	118·8	168·9

TABLE 5.—PRESSURE OF WATER AT DIFFERENT HEADS IN POUNDS PER SQUARE INCH (*Continued*).

Head of water in feet.	Head of water in yards.	Head of water in fathoms.	Head of water in metres.	Pressure of water in lbs. per square inch.	Head of water in feet.	Head of water in yards.	Head of water in fathoms.	Head of water in metres.	Pressure of water in lbs. per square inch.
395	131·6	65·8	120·3	171·1	650	216·6	108·3	198·1	281·6
400	133·3	66·6	121·9	173·3	660	220·0	110·0	201·1	285·9
405	135·0	67·5	123·4	175·4	670	223·3	111·6	204·2	290·3
410	136·6	68·3	124·9	177·6	680	226·6	113·3	207·2	294·6
415	138·3	69·1	126·4	179·8	690	230·0	115·0	210·3	298·9
420	140·0	70·0	128·0	181·9	700	233·3	116·6	213·3	303·3
425	141·6	70·8	129·5	184·1	710	236·6	118·3	216·4	307·6
430	143·3	71·6	131·0	186·3	720	240·0	120·0	219·4	311·9
435	145·0	72·5	132·5	188·4	730	243·3	121·6	222·5	316·3
440	146·6	73·3	134·1	190·6	740	246·6	123·3	225·5	320·6
445	148·3	74·1	135·6	192·8	750	250·0	125·0	228·6	324·9
450	150·0	75·0	137·1	194·9	760	253·3	126·6	231·6	329·3
455	151·6	75·8	138·6	197·1	770	256·6	128·3	234·6	333·6
460	153·3	76·6	140·2	199·3	780	260·0	130·0	237·7	337·9
465	155·0	77·5	141·7	201·4	790	263·3	131·6	240·7	342·3
470	156·6	78·3	143·2	203·6	800	266·6	133·3	243·8	346·6
475	158·3	79·1	144·7	205·8	810	270·0	135·0	246·8	350·9
480	160·0	80·0	146·3	207·9	820	273·3	136·6	249·9	355·3
485	161·6	80·8	147·8	210·1	830	276·6	138·3	252·9	359·6
490	163·3	81·6	149·3	212·3	840	280·0	140·0	256·0	363·9
495	165·0	82·5	150·8	214·4	850	283·3	141·6	259·0	368·3
500	166·6	83·3	152·4	216·6	860	286·6	143·3	262·1	372·6
510	170·0	85·0	155·4	220·9	870	290·0	145·0	265·1	376·9
520	173·3	86·6	158·4	225·3	880	293·3	146·6	268·2	381·3
530	176·6	88·3	161·5	229·6	890	296·6	148·3	271·2	385·6
540	180·0	90·0	164·5	233·9	900	300·0	150·0	274·3	389·9
550	183·3	91·6	167·6	238·3	910	303·3	151·6	277·3	394·3
560	186·6	93·3	170·6	242·6	920	306·6	153·3	280·4	398·6
570	190·0	95·0	173·7	246·9	930	310·0	155·0	283·4	402·9
580	193·3	96·6	176·7	251·3	940	313·3	156·6	286·5	407·3
590	196·6	98·3	179·8	255·6	950	316·6	158·3	289·5	411·6
600	200·0	100·0	182·8	259·9	960	320·0	160·0	292·6	415·9
610	203·3	101·6	185·9	264·3	970	323·3	161·6	295·6	420·3
620	206·6	103·3	188·9	268·6	980	326·6	163·3	298·7	424·6
630	210·0	105·0	192·0	272·9	990	330·0	165·0	301·7	428·9
640	213·3	106·6	195·0	277·3	1000	333·3	166·6	304·8	433·3

TABLE 6.—THEORETICAL DISCHARGE OF CIRCULAR ORIFICES OR NOZZLES IN ENGLISH GALLONS.

Head.		Velocity d'sch'ge,	Number of English gallons discharged per minute.																	
lbs.	feet.	ft. per ′.	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{4}$	$2\frac{1}{2}$
10	23·1	38·58	0·308	1·233	2·750	4·916	11·0	19·6	30·6	44·3	60·1	78·6	99·1	123	148	183	240	315	398	491
15	34·7	47·25	0·375	1·508	3·350	6·025	13·5	23·9	37·5	54·2	73·6	96·6	121	150	181	216	295	385	488	602
20	46·2	54·55	0·433	1·741	3·888	6·958	15·5	27·8	43·3	62·7	85·0	111	140	174	210	250	340	445	563	695
25	57·8	60·99	0·483	1·941	4·358	7·775	17·5	31·0	48·5	70·0	95·0	120	157	194	235	280	380	497	630	775
30	69·3	66·82	0·533	2·133	4·758	8·500	19·0	34·0	53·0	76·8	104	136	172	213	257	306	417	545	690	851
35	80·9	72·16	0·575	2·300	5·133	9·166	20·5	36·8	57·3	83·0	112	147	185	230	278	331	450	589	745	920
40	92·4	77·14	0·616	2·458	5·500	9·833	22·0	39·3	61·3	88·3	120	157	199	245	297	354	481	629	796	983
45	104·0	81·83	0·650	2·608	5·825	10·4	22·3	41·8	65·1	94·1	127	166	210	260	315	375	510	667	845	1043
50	115·5	86·26	0·683	2·750	6·141	11·0	24·5	44·0	68·5	99·1	134	175	222	275	332	395	538	704	890	1100
55	127·1	90·46	0·716	2·883	6·441	11·5	25·7	46·1	71·9	104	140	184	233	288	348	415	565	738	935	1150
60	138·6	94·49	0·750	3·016	6·733	12·0	26·9	48·1	74·5	108	147	192	244	301	364	433	590	770	975	1200
65	150 2	98·35	0·783	3·141	7·000	12·5	28·0	50·1	78·1	113	153	200	254	314	379	451	614	802	1015	1255
70	161·7	102·06	0·808	3·258	7·275	13·0	29·5	52·0	82·8	117	159	208	263	325	393	478	637	832	1054	1300
75	173·3	105·65	0·842	3·366	7·525	13·5	30·0	53·8	84·1	121	165	215	272	336	406	485	660	861	1090	1346
80	184·8	109·11	0·866	3·488	7·777	13·9	31·5	55·5	86·6	125	170	222	281	348	420	500	681	890	1126	1390
85	196·4	112·46	0·891	3·591	8·016	14·3	32·0	57·3	89·1	129	175	229	290	359	433	516	702	917	1161	1433
90	207·9	115·72	0·916	3·691	8·241	14·7	33·0	59·0	91·6	133	180	235	298	369	445	530	722	944	1195	1475
95	219·5	119·89	0·941	3·791	8·500	15·1	33·9	60·6	94·1	136	186	242	306	379	458	545	742	970	1228	1516
100	231·1	121·98	0·966	3·891	8·666	15·5	34·7	62·1	96·5	140	190	249	315	389	470	560	761	995	1260	1555
105	242·6	125·00	0·991	3·983	8·916	15·9	35·6	63·7	99·1	143	195	255	322	398	481	573	780	1020	1290	1593
110	254·2	127·94	1·016	4·083	9·083	16·3	36·5	65·2	101	147	199	260	330	408	492	587	799	1044	1321	1630
115	265·7	130·82	1·041	4·175	9·333	16·6	37·3	66·7	104	150	204	266	337	417	504	590	816	1067	1350	1666
120	277·3	133·63	1·059	4·266	9·500	17·0	38·0	68·1	105	153	208	272	345	426	515	613	834	1090	1380	1703
125	288·8	136·38	1·083	4·350	9·750	17·4	38·9	69·5	108	156	212	279	351	436	525	625	851	1112	1409	1738
130	300·4	139·08	1·108	4·433	9·916	17·7	39·6	70·9	110	160	216	284	359	443	535	638	868	1136	1436	1773

TABLE 7.—THEORETICAL DISCHARGE OF CIRCULAR ORIFICES OR NOZZLES IN UNITED STATES GALLONS.

Head.		Velocity d'sch'ge,	Number of United States gallons discharged per minute.																	
lbs.	feet.	ft. per ″.	1/16	1/8	3/16	1/4	3/8	1/2	5/8	3/4	7/8	1	1⅛	1¼	1⅜	1½	1¾	2	2¼	2½
10	23·1	38·58	0·37	1·48	3·30	5·90	13·2	23·6	36·8	53·2	72·2	94·4	119	148	178	212	289	378	478	590
15	34·7	47·25	0·45	1·81	4·02	7·23	16·2	28·7	45·0	65·1	88·4	116	146	181	218	260	354	463	586	723
20	46·2	54·55	0·52	2·09	4·66	8·35	18·7	33·4	52·0	75·3	102	134	169	209	252	300	409	534	676	835
25	57·8	60·99	0·58	2·33	5·23	9·33	20·9	37·2	58·2	84·1	114	149	189	233	282	336	457	597	756	933
30	69·3	66·82	0·64	2·56	5·71	10·2	22·8	40·9	63·7	92·2	125	164	207	256	309	368	501	654	828	1022
35	80·9	72·16	0·69	2·76	6·16	11·0	24·7	44·2	68·8	99·6	135	177	223	276	334	397	541	707	895	1104
40	92·4	77·14	0·74	2·95	6·60	11·8	26·4	47·2	73·6	106	144	189	239	295	357	425	578	755	956	1180
45	104·0	81·83	0·78	3·13	6·99	12·5	28·0	50·2	78·1	113	153	200	253	313	378	450	613	801	1014	1252
50	115·5	86·26	0·82	3·30	7·37	13·2	29·5	52·8	82·3	119	161	211	267	330	399	475	646	845	1069	1320
55	127·1	90·46	0·86	3·46	7·73	13·8	30·9	55·4	86·3	125	169	221	280	346	418	498	678	886	1122	1385
60	138·6	94·49	0·90	3·62	8·08	14·5	32·3	57·8	90·1	130	177	231	293	362	437	520	708	925	1171	1446
65	150·2	98·35	0·94	3·77	8·40	15·1	33·6	60·2	93·8	136	184	241	305	377	455	542	737	963	1219	1506
70	161·7	102·06	0·97	3·91	8·73	15·6	34·9	62·5	97·4	141	191	250	316	391	472	562	765	999	1265	1561
75	173·3	105·65	1·01	4·04	9·03	16·2	36·1	64·6	101	146	198	259	327	494	488	582	792	1034	1309	1616
80	184·8	109·11	1·04	4·18	9·33	16·7	37·8	66·6	104	150	204	267	338	418	504	601	818	1068	1352	1669
85	196·4	112·46	1·07	4·31	9·62	17·2	38·5	68·8	107	155	210	275	348	431	520	620	843	1101	1394	1720
90	207·9	115·72	1·10	4·43	9·89	17·7	39·6	70·8	110	160	217	283	358	443	535	637	867	1133	1434	1770
95	219·5	118·89	1·13	4·55	10·2	18·2	40·7	72·8	113	164	223	291	368	455	550	655	891	1164	1474	1820
100	231·1	121·98	1·16	4·67	10·4	18·7	41·7	74·6	116	168	228	299	378	467	564	672	914	1194	1512	1866
105	242·6	125·00	1·19	4·78	10·7	19·1	42·8	76·5	119	172	234	306	387	478	578	688	937	1224	1549	1912
110	254·2	127·94	1·22	4·90	10·9	19·6	43·8	78·3	122	177	239	313	396	490	591	705	959	1253	1586	1957
115	265·7	130·82	1·25	5·01	11·2	20·0	44·8	80·1	125	181	245	320	405	501	605	720	980	1281	1621	2002
120	277·3	133·63	1·27	5·12	11·4	20·4	45·7	81·8	127	184	250	327	414	512	618	736	1001	1308	1656	2044
125	288·8	136·38	1·30	5·22	11·7	20·9	46·7	83·5	130	188	255	334	422	522	630	751	1022	1335	1691	2086
130	300·4	139·08	1·33	5·32	11·9	21·3	47·6	85·1	133	192	260	341	431	532	643	766	1042	1362	1724	2128

The actual discharge in these two tables will be for ring nozzles = 64 per cent.; and for tapering smooth nozzles = 82 per cent

Velocity of flow of water in pipes (Kütter's rules).

For smooth pipes.

Pipes ½ inch to 2½ inches diameter = $V = 107\ D^{0\cdot9}\ \sqrt{S}$;
,, 2½ ins. to 5 ,, ,, = $V = 115\ D^{0\cdot8}\ \sqrt{S}$;
,, 5 ,, 10 ,, ,, = $V = 134\ D^{0\cdot7}\ \sqrt{S}$;
,, 10 ,, 72 ,, ,, = $V = 166\ D^{0\cdot6}\ \sqrt{S}$;
,, 6 feet to 400 feet ,, = $V = 256\ \sqrt{DS}$;

When V = velocity in feet per second;
D = diameter of pipe in inches;
S = hydraulic inclination.

For moderately smooth pipes.

Pipes ½ inch to 2½ inches diameter = $V = 63\ D\ \sqrt{S}$;
,, 2½ ins. to 5 ,, ,, = $V = 68\ D^{0\cdot9}\ \sqrt{S}$;
,, 5 ,, 10 ,, ,, = $V = 78\ D^{0\cdot8}\ \sqrt{S}$;
,, 10 ,, 24 ,, ,, = $V = 100\ D^{0\cdot7}\ \sqrt{S}$;
,, 24 ,, 96 ,, ,, = $V = 138\ D^{0\cdot6}\ \sqrt{S}$;
,, 8 feet to 400 feet ,, = $V = 221\ \sqrt{DS}$.

Usual inclination of pipes.

1 inch in 12 feet = minimum fall for house drains;
1 ,, ,, 16 ,, = ,, ,, ,, land drains;
1 ,, ,, 40 ,, = ,, ,, ,, sub-drains for houses;
1 ,, ,, 100 ,, = ,, ,, ,, main drains for houses;
1 ,, ,, 150 ,, = fall of mountain torrents;
1 ,, ,, 230 ,, = ,, ,, rivers and rapid currents;
1 ,, ,, 280 ,, = ,, ,, strong currents;
1 ,, ,, 340 ,, = ,, ,, ordinary rivers with good current;
1 ,, ,, 440 ,, = ,, ,, winding rivers subject to inundations with slow current;
1 ,, ,, 480 ,, = ,, ,, water channels, supply pipes to reservoirs and small canals;
1 ,, ,, 570 ,, = ,, ,, large canals;
1 ,, ,, 1000 ,, = very slow current, nearly approaching to stagnant water.

TABLE 8.—FRICTION OF WATER IN PIPES 100 FEET IN LENGTH.

Internal diameter of pipe in inches.										
		3			3½			4		
Velocity of water in feet per second.	Head to produce this velocity in feet.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.
2·0	·062	·659	36·7	5·89	·565	50·0	8·02	·494	64·8	10·4
2·2	·075	·780	40·4	6·48	·669	55·0	8·82	·585	71·7	11·5
2·4	·090	·911	44·1	7·07	·781	60·0	9·62	·683	78·0	12·5
2·6	·105	1·05	47·7	7·65	·901	64·8	10·4	·788	84·8	13·6
2·8	·122	1·20	52·4	8·24	1·03	69·8	11·2	·900	91·1	14·6
3·0	·140	1·35	55·0	8·83	1·16	74·8	12·0	1·02	97·9	15·7
3·2	·160	1·52	58·7	9·42	1·31	79·8	12·8	1·14	104	16·7
3·4	·180	1·70	62·4	10·0	1·46	84·8	13·6	1·27	111	17·8
3·6	·202	1·89	66·1	10·6	1·62	89·8	14·4	1·41	117	18·8
3·8	·225	2·08	69·8	11·2	1·78	94·8	15·2	1·56	124	19·9
4·0	·250	2·28	73·6	11·8	1·96	99·8	16·0	1·71	130	20·9
4·2	·275	2·49	76·7	12·3	2·14	104	16·8	1·87	137	22·0
4·4	·302	2·71	80·4	12·9	2·33	109	17·6	2·03	143	23·0
4·6	·330	2·94	84·2	13·5	2·52	114	18·4	2·21	149	24·0
4·8	·360	3·18	87·9	14·1	2·72	119	19·2	2·38	156	25·1
5·0	·390	3·43	91·7	14·7	2·94	124	20·0	2·57	163	26·2
5·2	·422	3·68	95·4	15·3	3·15	129	20·8	2·76	169	27·2
5·4	·455	3·94	99·2	15·9	3·38	134	21·6	2·96	175	28·2
5·6	·490	4·22	102	16·5	3·61	139	22·4	3·16	182	29·3
5·8	·525	4·50	106	17·1	3·85	144	23·2	3·37	189	30·3
6·0	·562	4·78	110	17·7	4·10	149	24·0	3·59	195	31·4
6·2	·600	5·08	113	18·2	4·36	154	24·8	3·81	202	32·4
6·4	·640	5·39	117	18·8	4·62	159	25·6	4·04	209	33·5
6·6	·680	5·70	121	19·4	4·89	164	26·4	4·28	215	34·5
6·8	·722	6·02	124	20·0	5·16	169	27·3	4·52	222	35·6
7·0	·765	6·35	128	20·6	5·45	174	28·0	4·77	228	36·6

TABLE 8 (*Continued*).

Internal diameter of pipe in inches.							
		4½			5		
Velocity of water in feet per second.	**Head to produce this velocity in feet.**	**Head to overcome friction in feet.**	**Gallons per minute.**	**Cubic feet per minute.**	**Head to overcome friction in feet.**	**Gallons per minute.**	**Cubic feet per minute.**
2·0	·062	·439	82·3	13·2	·395	101	16·3
2·2	·075	·520	91·1	14·6	·468	112	18·0
2·4	·090	·607	99·2	15·9	·547	122	19·6
2·6	·105	·701	107	17·2	·631	132	21·3
2·8	·122	·800	115	18·5	·720	142	22·9
3·0	·140	·905	123	19·8	·815	152	24·5
3·2	·160	1·02	132	21·2	·915	163	26·2
3·4	·180	1·13	140	22·5	1·02	173	27·8
3·6	·202	1·26	148	23·8	1·13	183	29·4
3·8	·225	1·39	157	25·2	1·25	193	31·0
4·0	·250	1·52	165	26·5	1·37	204	32·7
4·2	·275	1·66	173	27·8	1·50	214	34·3
4·4	·302	1·81	181	29·1	1·63	224	36·0
4·6	·330	1·96	189	30·4	1·76	234	37·6
4·8	·360	2·12	198	31·8	1·91	244	39·2
5·0	·390	2·28	206	33·1	2·05	255	40·9
5·2	·422	2·45	214	34·4	2·21	265	42·5
5·4	·455	2·63	223	35·8	2·37	275	44·2
5·6	·490	2·81	231	37·1	2·53	285	45·8
5·8	·525	3·00	239	38·4	2·70	295	47·4
6·0	·562	3·19	247	39·7	2·87	305	49·0
6·2	·600	3·39	255	41·0	3·05	316	50·7
6·4	·640	3·59	264	42·4	3·23	326	52·3
6·6	·680	3·80	272	43·7	3·42	336	54·0
6·8	·722	4·01	280	45·0	3·61	346	55·6
7·0	·765	4·24	289	46·4	3·81	356	57·2

TABLE 8 (*Continued*).

Internal diameter of pipe in inches.										
		6			7			8		
Velocity of water in feet per second.	Head to produce this velocity in feet.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.
2·0	·062	·329	146	23·5	·282	199	32·0	·247	261	41·9
2·2	·075	·390	161	25·9	·334	220	35·3	·293	287	46·1
2·4	·090	·456	175	28·2	·390	240	38·5	·342	313	50·2
2·6	·105	·526	190	30·6	·450	260	41·7	·394	339	54·4
2·8	·122	·600	205	32·9	·514	280	44·9	·450	365	58·6
3·0	·140	·679	220	35·3	·582	300	48·1	·509	391	62·8
3·2	·160	·763	235	37·7	·654	320	51·3	·572	418	67·0
3·4	·180	·851	249	40·0	·729	340	54·5	·638	444	71·2
3·6	·202	·943	264	42·4	·808	360	57·7	·707	470	75·4
3·8	·225	1·04	278	44·7	·892	380	60·9	·780	496	79·6
4·0	·250	1·14	293	47·1	·979	399	64·1	·856	522	83·7
4·2	·275	1·25	308	49·5	1·07	419	67·3	·935	548	87·9
4·4	·302	1·35	313	51·8	1·16	439	70·5	1·02	574	92·1
4·6	·330	1·47	337	54·1	1·26	459	73·7	1·10	600	96·3
4·8	·360	1·59	352	56·5	1·36	479	76·9	1·19	624	100
5·0	·390	1·71	367	58·9	1·47	499	80·2	1·28	655	105
5·2	·422	1·84	381	61·2	1·58	519	83·3	1·38	680	109
5·4	·455	1·97	396	63·6	1·69	540	86·6	1·48	705	113
5·6	·490	2·11	411	65·9	1·81	560	89·8	1·58	730	117
5·8	·525	2·25	426	68·3	1·93	580	93·0	1·68	755	121
6·0	·562	2·39	441	70·7	2·05	600	96·2	1·79	780	125
6·2	·600	2·54	455	73·0	2·18	620	99·4	1·90	811	130
6·4	·640	2·69	470	75·4	2·31	636	102	2·02	836	134
6·6	·680	2·85	484	77·7	2·44	661	106	2·14	861	138
6·8	·722	3·01	499	80·1	2·58	680	109	2·26	886	142
7·0	·765	3·18	514	82·4	2·72	698	112	2·38	911	146

TABLE 8 (*Continued*).

Internal diameter of pipe in inches.							
		9			10		
Velocity of water in feet per second.	Head to produce this velocity in feet.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.
2·0	·062	·220	330	53·0	·198	408	65·4
2·2	·075	·260	363	58·3	·234	449	72·0
2·4	·090	·304	396	63·6	·273	489	78·5
2·6	·105	·350	429	68·9	·315	531	85·1
2·8	·122	·400	463	74·2	·360	571	91·6
3·0	·140	·453	496	79·5	·407	612	98·2
3·2	·160	·508	529	84·8	·458	655	105
3·4	·180	·567	562	90·1	·510	692	111
3·6	·202	·629	595	95·4	·566	736	118
3·8	·225	·693	630	101	·624	773	124
4·0	·250	·761	661	106	·685	817	131
4·2	·275	·832	692	111	·748	854	137
4·4	·302	·905	723	116	·814	898	144
4·6	·330	·981	761	122	·883	936	150
4·8	·360	1·06	792	127	·954	979	157
5·0	·390	1·14	823	132	1·03	1017	163
5·2	·422	1·23	861	138	1·10	1060	170
5·4	·455	1·31	892	143	1·18	1104	177
5·6	·490	1·40	923	148	1·26	1141	183
5·8	·525	1·50	960	154	1·35	1185	190
6·0	·562	1·59	992	159	1·43	1223	196
6·2	·600	1·69	1023	164	1·52	1266	203
6·4	·640	1·79	1054	169	1·61	1304	209
6·6	·680	1·90	1092	175	1·71	1347	216
6·8	·722	2·01	1123	180	1·81	1385	222
7·0	·765	2·12	1144	185	1·90	1428	229

TABLE 8 (*Continued*).

		Internal diameter of pipe in inches.								
		11			12			13		
Velocity of water in feet per second.	Head to produce this velocity in feet.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.
2·0	·062	·180	494	72·2	·165	587	94·2	·152	686	110
2·2	·075	·213	543	87·1	·195	642	103	·180	755	121
2·4	·090	·248	592	95·0	·228	705	113	·210	829	133
2·6	·105	·287	642	103	·263	761	122	·242	898	144
2·8	·122	·327	692	111	·300	823	132	·277	973	156
3·0	·140	·370	742	119	·339	879	141	·313	1035	166
3·2	·160	·416	792	127	·381	942	151	·352	1104	177
3·4	·180	·464	836	134	·425	998	160	·393	1173	188
3·6	·202	·514	886	142	·472	1054	169	·435	1241	199
3·8	·225	·567	936	150	·520	1116	179	·480	1310	210
4·0	·250	·623	985	158	·571	1173	188	·527	1379	221
4·2	·275	·680	1035	166	·624	1235	198	·576	1447	232
4·4	·302	·740	1085	174	·679	1291	207	·626	1516	243
4·6	·330	·803	1135	182	·736	1354	217	·679	1584	254
4·8	·360	·867	1185	190	·795	1410	226	·734	1653	265
5·0	·390	·935	1235	198	·857	1466	235	·791	1722	276
5·2	·422	1·00	1285	206	·920	1528	245	·850	1790	287
5·4	·455	1·07	1335	214	·986	1584	254	·910	1859	298
5·6	·490	1·15	1385	222	1·05	1647	264	·973	1928	309
5·8	·525	1·22	1428	229	1·12	1703	273	1·04	2003	321
6·0	·562	1·30	1478	237	1·19	1765	283	1·10	2071	332
6·2	·600	1·38	1528	245	1·27	1822	292	1·17	2140	343
6·4	·640	1·47	1578	253	1·35	1878	301	1·24	2208	354
6·6	·680	1·55	1628	261	1·42	1940	311	1·31	2277	365
6·8	·722	1·64	1678	269	1·50	1996	320	1·39	2346	376
7·0	·765	1·73	1728	277	1·59	2059	330	1·46	2414	387

TABLE 8 (*Continued*).

Internal diameter of pipe in inches.							
		14			15		
Velocity of water in feet per second.	Head to produce this velocity in feet.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.
2·0	·062	·141	798	128	·132	917	147
2·2	·075	·167	879	141	·156	1010	162
2·4	·090	·195	950	154	·182	1098	176
2·6	·105	·225	1042	167	·210	1191	191
2·8	·122	·257	1116	179	·240	1285	206
3·0	·140	·291	1235	198	·271	1379	221
3·2	·160	·327	1279	205	·305	1466	235
3·4	·180	·365	1360	218	·340	1560	250
3·6	·202	·404	1441	231	·377	1653	265
3·8	·225	·446	1516	243	·416	1747	280
4·0	·250	·489	1597	256	·457	1834	294
4·2	·275	·534	1678	269	·499	1928	309
4·4	·302	·582	1759	282	·543	2021	324
4·6	·330	·634	1830	295	·589	2115	339
4·8	·360	·682	1921	308	·636	2202	353
5·0	·390	·734	2003	321	·685	2296	368
5·2	·422	·789	2077	333	·736	2389	383
5·4	·455	·845	2159	346	·789	2471	397
5·6	·490	·903	2240	359	·843	2570	412
5·8	·525	·964	2321	372	·899	2664	427
6·0	·562	1·02	2402	385	·957	2758	442
6·2	·600	1·09	2477	397	1·01	2845	456
6·4	·640	1·15	2558	410	1·08	2939	471
6·6	·680	1·22	2639	423	1·14	3032	486
6·8	·722	1·29	2720	436	1·20	3120	500
7·0	·765	1·36	2801	449	1·27	3213	515

TABLE 8 (*Continued*).

		Internal diameter of pipe in inches.								
		16			18			20		
Velocity of water in feet per second.	Head to produce this velocity in feet.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.
2·0	·062	·123	1042	167	·110	1322	212	·099	1634	262
2·2	·075	·146	1148	184	·130	1493	233	·117	1797	288
2·4	·090	·171	1254	201	·152	1584	254	·137	1959	314
2·6	·105	·197	1360	218	·175	1716	275	·158	2121	340
2·8	·122	·225	1460	234	·200	1853	297	·180	2283	366
3·0	·140	·255	1566	251	·226	1984	318	·204	2452	393
3·2	·160	·286	1672	268	·254	2115	339	·229	2614	419
3·4	·180	·319	1772	284	·283	2246	360	·255	2776	445
3·6	·202	·354	1878	301	·314	2383	382	·283	2939	471
3·8	·225	·390	1984	318	·347	2514	403	·312	3101	497
4·0	·250	·428	2090	335	·380	2645	424	·342	3263	523
4·2	·275	·468	2196	352	·416	2776	445	·374	3432	550
4·4	·302	·509	2296	368	·452	2907	466	·407	3594	576
4·6	·330	·552	2402	385	·490	3045	488	·441	3756	602
4·8	·360	·596	2508	402	·530	3176	509	·477	3918	628
5·0	·390	·642	2614	419	·571	3307	530	·514	4080	654
5·2	·422	·690	2714	435	·614	3438	551	·552	4243	680
5·4	·455	·740	2820	452	·657	3569	572	·592	4411	707
5·6	·490	·791	2926	469	·703	3706	594	·632	4573	733
5·8	·525	·843	3032	486	·749	3837	615	·674	4736	759
6·0	·562	·897	3132	502	·798	3968	636	·718	4898	785
6·2	·600	·953	3238	519	·847	4099	657	·762	5060	811
6·4	·640	1·01	3344	536	·898	4230	678	·808	5229	838
6·6	·680	1·07	3450	553	·950	4368	700	·855	5391	864
6·8	·722	1·13	3550	569	1·00	4499	721	·904	5553	890
7·0	·765	1·19	3656	586	1·06	4630	742	·953	5715	916

TABLE 8 (*Continued*).

		Internal diameter of pipe in inches.					
		22			24		
Velocity of water in feet per second.	Head to produce this velocity in feet.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.
2·0	·062	·090	1971	316	·082	2352	377
2·2	·075	·106	2171	348	·097	2583	414
2·4	·090	·124	2371	380	·114	2820	452
2·6	·105	·143	2570	412	·131	3057	490
2·8	·122	·164	2764	443	·150	3294	528
3·0	·140	·185	2964	475	·170	3525	565
3·2	·160	·208	3163	507	·191	3762	603
3·4	·180	·232	3357	538	·213	3999	641
3·6	·202	·257	3556	570	·236	4230	678
3·8	·225	·284	3750	601	·260	4467	716
4·0	·250	·311	3949	633	·285	4704	754
4·2	·275	·340	4149	665	·312	4935	791
4·4	·302	·370	4349	697	·339	5172	829
4·6	·330	·401	4542	728	·368	5410	867
4·8	·360	·434	4742	760	·397	5647	905
5·0	·390	·467	4942	792	·428	5878	942
5·2	·422	·502	5135	823	·460	6075	980
5·4	·455	·538	5335	855	·493	6352	1018
5·6	·490	·575	5534	887	·527	6583	1055
5·8	·525	·613	5728	918	·562	6820	1093
6·0	·562	·652	5928	950	·598	7057	1131
6·2	·600	·693	6127	982	·635	7288	1168
6·4	·640	·735	6321	1013	·673	7525	1206
6·6	·680	·778	6520	1045	·713	7762	1244
6·8	·722	·821	6720	1077	·753	7999	1282
7·0	·765	·867	6920	1109	·794	8230	1319

TABLE 8 (*Continued*).

		Internal diameter of pipe in inches.								
		26			28			30		
Velocity of water in feet per second.	Head to produce this velocity in feet.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.	Head to overcome friction in feet.	Gallons per minute.	Cubic feet per minute.
2·0	·062	·076	2758	442	·070	3201	513	·066	3675	589
2·2	·075	·090	3032	486	·083	3519	564	·078	4043	684
2·4	·090	·105	3313	531	·097	3843	616	·091	4411	707
2·6	·105	·121	3588	575	·112	4162	667	·105	4789	766
2·8	·122	·138	3862	619	·128	4480	718	·120	5141	824
3·0	·140	·157	4137	663	·145	4804	770	·136	5509	883
3·2	·160	·176	4417	708	·163	5123	821	·152	5878	942
3·4	·180	·196	4692	752	·182	5441	872	·170	6246	1001
3·6	·202	·218	4967	796	·202	5759	923	·189	6614	1060
3·8	·225	·240	5241	840	·223	6077	974	·208	6982	1119
4·0	·250	·263	5522	885	·244	6402	1026	·228	7350	1178
4·2	·275	·288	5796	929	·267	6720	1077	·249	7718	1237
4·4	·302	·313	6071	973	·290	7044	1129	·271	8087	1296
4·6	·330	·339	6346	1017	·315	7363	1180	·294	8455	1355
4·8	·360	·367	6626	1062	·341	7681	1231	·318	8823	1414
5·0	·390	·395	6901	1106	·367	8005	1283	·343	9185	1472
5·2	·422	·425	7176	1150	·394	8324	1334	·368	9553	1531
5·4	·455	·455	7450	1194	·423	8642	1385	·394	9921	1590
5·6	·490	·486	7731	1239	·452	8966	1437	·422	10289	1649
5·8	·525	·519	8005	1283	·482	9285	1488	·450	10657	1708
6·0	·562	·552	8280	1327	·513	9603	1539	·478	11026	1767
6·2	·600	·586	8555	1371	·544	9921	1590	·508	11394	1826
6·4	·640	·622	8835	1416	·577	10246	1642	·539	11762	1885
6·6	·680	·658	9110	1460	·611	10564	1693	·570	12124	1943
6·8	·722	·695	9381	1504	·645	10882	1744	·602	12498	2003
7·0	·765	·733	9659	1548	·681	11207	1796	·635	12860	2061

TABLE 9.—SOCKET PIPES (*not turned and bored.*)
(FIG. 1.)

A inch.	B inch.	C inch.	D inch.	E inch.	F inch.	G inch.	H inch.	J inch.	K inch.	L inch.
2	3	3	3¼	5	¾	11/16	⅝	¾	3	2¾
3	3	3	4¼	6	¾	11/16	⅝	¾	4	3¾
4	3	3	5¼	7	¾	11/16	⅝	¾	5	4¾
5	3½	3½	6⅜	8⅝	⅞	13/16	11/16	⅞	6¼	5⅞
6	3½	3½	7⅝	9⅝	⅞	13/16	11/16	⅞	7¼	6⅞
7	3½	3½	8¾	10¾	⅞	13/16	11/16	⅞	8¼	8
8	4	4	9¾	12	1	13/16	11/16	⅞	9¼	9
9	4	4	10⅞	13⅛	1	13/16	11/16	⅞	10½	10⅛
12	4	4	14	16½	1	15/16	13/16	1⅛	13½	13¼
15	4½	4½	17⅛	19⅞	1⅛	1 1/16	15/16	1¼	16¾	16⅜
18	4½	4½	20⅜	23⅜	1¼	1⅛	1	1¼	20	19⅝
20	4½	4½	22½	25¾	1⅜	1¼	1⅛	1¼	22¼	21¾
24	5	5	26¾	30¼	1½	1⅜	1¼	1½	26¼	26
35	5½	5	35¾	37¾	1¾	1½	1⅜	1⅝	35⅝	35

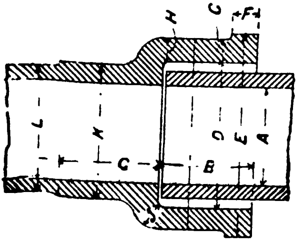


FIG. 1.—SOCKET PIPES (*not turned and bored*).

TABLE 10.—SOCKET PIPES (*turned and bored.*)
(FIG 2.)

A inch.	B inch.	C inch.	D inch.	E inch.	F inch.	G inch.	H inch.	J inch.	K inch.	L inch.	M inch.
2	3	3 1/4	3 1/4	5	3/4	11/16	5/8	3/4	3	1 1/4	2 3/4
3	3	3	4 1/4	6	3/4	11/16	5/8	3/4	4	1 1/4	3 3/4
4	3	3	5 1/4	7	3/4	11/16	5/8	3/4	5	1 1/4	4 3/4
5	3 1/2	3 1/2	6 5/8	8 5/8	7/8	13/16	11/16	7/8	6 1/4	1 3/8	5 7/8
6	3 1/2	3 1/2	7 5/8	9 5/8	7/8	13/16	11/16	7/8	7 1/4	1 3/8	6 7/8
7	3 1/2	3 1/2	8 3/4	10 3/4	7/8	13/16	11/16	7/8	8 1/4	1 3/8	8
8	4	4	9 3/4	12	1	13/16	11/16	7/8	9 1/4	1 5/8	9
9	4	4	10 7/8	13 1/8	1	13/16	11/16	7/8	10 1/2	1 5/8	10 1/8
12	4 1/2	4 1/2	14	16 1/2	1	15/16	13/16	1	13 1/2	1 3/4	13 1/4

The taper of the bored portions is $\frac{1}{32}$ inch per inch length.

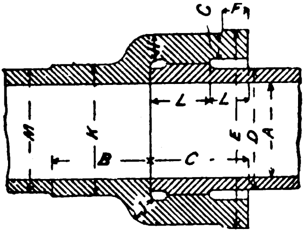


FIG. 2.—SOCKET PIPES (*turned and bored*).

TABLE 11.—THICKNESS OF PIPES REQUIRED FOR DIFFERENT HEADS OF WATER IN FEET. STRESS ON METAL BEING TAKEN AT 2,500 LBS. PER SQUARE INCH.

Head of Water in Feet.	Diameter of Pipes in Inches.																								
	2	2½	3	3½	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	Thickness of Metal in Inches and Fractions of an Inch.																								
10	¼	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜
20	¼	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜
30	¼	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜
40	¼	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜
50	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜
60	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜
70	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜
80	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜
90	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½
100	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½
110	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	½	½
120	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	½	⅝	⅝
130	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	½	½	⅝	⅝	⅝
140	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	½	⅝	⅝	⅝	⅝	⅝
150	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	½	⅝	⅝	⅝	⅝	¾	¾
160	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	½	⅝	⅝	⅝	⅝	¾	¾	¾
170	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	⅝	⅝	⅝	¾	¾	¾	¾	⅞
180	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	⅝	⅝	⅝	¾	¾	¾	¾	⅞	⅞
190	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	⅝	⅝	⅝	¾	¾	¾	¾	⅞	⅞	⅞
200	¼	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	⅝	⅝	¾	¾	¾	¾	⅞	⅞	⅞	1
210	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	½	⅝	⅝	⅝	¾	¾	¾	⅞	⅞	⅞	1	1
220	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	⅝	⅝	¾	¾	¾	⅞	⅞	⅞	1	1	1
230	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	⅝	⅝	⅝	¾	¾	¾	⅞	⅞	⅞	1	1	1	1⅛
240	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	⅝	⅝	¾	¾	⅞	⅞	⅞	1	1	1	1⅛	1⅛
250	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	⅝	⅝	⅝	¾	¾	¾	⅞	⅞	1	1	1	1⅛	1⅛	1⅛
260	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	⅝	⅝	¾	¾	⅞	⅞	⅞	1	1	1⅛	1⅛	1⅛	1¼
270	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	⅝	¾	¾	¾	⅞	⅞	1	1	1	1⅛	1⅛	1¼	1¼
280	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	½	½	⅝	⅝	⅝	¾	¾	⅞	⅞	1	1	1	1⅛				
290	¼	¼	⅜	⅜	⅜	⅜	⅜	⅜	½	½	⅝	⅝	¾	¾	¾	⅞	⅞	1	1						
300	¼	⅜	⅜	⅜	⅜	⅜	⅜	⅜	½	½	⅝	⅝	¾	¾	⅞	⅞	1	1	1						

TABLE 11 (*Continued*).

Head of Water in Feet.	Diameter of Pipes in Inches.																								
	2	2½	3	3½	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
	Thickness of Metal in Inches and Fractions of an Inch.																								
310	¼	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	⅝	¾	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1¼	1¼	1¼	1⅜
320	¼	⅜	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	⅝	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1⅛	1¼	1¼	1⅜	1⅜
330	¼	⅜	⅜	⅜	⅜	⅜	⅜	½	½	⅝	⅝	¾	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1¼	1¼	1⅜	1⅜	1⅜
340	¼	⅜	⅜	⅜	⅜	⅜	⅜	½	½	⅝	⅝	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1⅛	1¼	1¼	1⅜	1⅜	1½
350	¼	⅜	⅜	⅜	⅜	⅜	⅜	½	½	⅝	⅝	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1⅛	1¼	1⅜	1⅜	1½	1½
360	¼	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	¾	¾	⅞	⅞	1	1	1	1⅛	1⅛	1¼	1¼	1⅜	1⅜	1½	1½
370	¼	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1¼	1¼	1⅜	1⅜	1½	1½	1⅝
380	¼	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1¼	1¼	1⅜	1⅜	1½	1⅝	1⅝
390	¼	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	¾	¾	⅞	1	1	1	1⅛	1¼	1¼	1⅜	1⅜	1⅜	1½	1⅝	1⅝
400	¼	⅜	⅜	⅜	⅜	⅜	½	½	½	⅝	¾	⅞	⅞	1	1	1⅛	1⅛	1¼	1¼	1⅜	1⅜	1½	1⅝	1⅝	1¾
410	¼	⅜	⅜	⅜	⅜	⅜	½	½	⅝	¾	¾	⅞	⅞	1	1	1⅛	1¼	1¼	1¼	1⅜	1½	1½	1⅝	1¾	1¾
420	¼	⅜	⅜	⅜	⅜	⅜	½	⅝	⅝	¾	¾	⅞	⅞	1	1⅛	1⅛	1¼	1¼	1⅜	1⅜	1½	1⅝	1⅝	1¾	1¾
430	¼	⅜	⅜	⅜	⅜	⅜	½	⅝	⅝	¾	¾	⅞	1	1	1⅛	1¼	1¼	1¼	1⅜	1½	1½	1⅝	1¾	1¾	1⅞
440	¼	⅜	⅜	⅜	⅜	½	½	⅝	⅝	¾	⅞	⅞	1	1	1⅛	1¼	1¼	1⅜	1⅜	1½	1⅝	1⅝	1¾	1⅞	1⅞
450	¼	⅜	⅜	⅜	⅜	½	½	⅝	⅝	¾	⅞	⅞	1	1⅛	1⅛	1¼	1¼	1⅜	1½	1½	1⅝	1¾	1¾	1⅞	1⅞
460	¼	⅜	⅜	⅜	⅜	½	½	⅝	¾	¾	⅞	⅞	1	1⅛	1⅛	1¼	1⅜	1⅜	1½	1½	1⅝	1¾	1⅞	1⅞	2
470	¼	⅜	⅜	⅜	⅜	½	½	⅝	¾	¾	⅞	⅞	1	1⅛	1¼	1¼	1⅜	1½	1½	1⅝	1¾	1¾	1⅞	1⅞	2
480	¼	⅜	⅜	⅜	⅜	½	½	⅝	¾	¾	⅞	1	1	1⅛	1¼	1¼	1⅜	1½	1½	1⅝	1¾	1¾	1⅞	1⅞	2
490	¼	⅜	⅜	⅜	⅜	½	⅝	⅝	¾	⅞	⅞	1	1⅛	1⅛	1¼	1⅜	1⅜	1½	1⅝	1⅝	1¾	1⅞	1⅞	2	2⅛
500	¼	⅜	⅜	⅜	⅜	½	⅝	⅝	¾	⅞	⅞	1	1⅛	1⅛	1¼	1⅜	1½	1½	1⅝	1¾	1¾	1⅞	2	2	2⅛
510	¼	⅜	⅜	⅜	⅜	½	⅝	⅝	¾	⅞	1	1	1⅛	1¼	1¼	1⅜	1½	1⅝	1⅝	1¾	1⅞	1⅞	2	2⅛	2⅛
520	¼	⅜	⅜	⅜	⅜	½	⅝	¾	¾	⅞	1	1	1⅛	1¼	1⅜	1⅜	1½	1⅝	1⅝	1¾	1⅞	2	2	2⅛	2¼
530	¼	⅜	⅜	⅜	⅜	½	⅝	¾	¾	⅞	1	1⅛	1⅛	1¼	1⅜	1⅜	1½	1⅝	1¾	1¾	1⅞	2	2⅛	2⅛	2¼
540	¼	⅜	⅜	⅜	½	½	⅝	¾	¾	⅞	1	1⅛	1⅛	1¼	1⅜	1½	1½	1⅝	1¾	1⅞	1⅞	2	2⅛	2¼	2¼
550	¼	⅜	⅜	⅜	½	½	⅝	¾	⅞	⅞	1	1⅛	1¼	1¼	1⅜	1½	1⅝	1⅝	1¾	1⅞	2	2	2⅛	2¼	2⅜
560	¼	⅜	⅜	⅜	½	½	⅝	¾	⅞	⅞	1	1⅛	1¼	1⅜	1⅜	1½	1⅝	1¾	1¾	1⅞	2	2⅛	2¼	2¼	2⅜
570	¼	⅜	⅜	⅜	½	½	⅝	¾	⅞	1	1	1⅛	1¼	1⅜	1½	1½	1⅝	1¾	1⅞	1⅞	2	2⅛	2¼	2⅜	2⅜
580	¼	⅜	⅜	⅜	½	⅝	⅝	¾	⅞	1	1⅛	1⅛	1¼	1⅜	1½	1⅝	1⅝	1¾	1⅞	2	2⅛	2⅛	2¼	2⅜	2½
590	¼	⅜	⅜	⅜	½	⅝	⅝	¾	⅞	1	1⅛	1¼	1¼	1⅜	1½	1⅝	1¾	1¾	1⅞	2	2⅛	2¼	2¼	2⅜	2½
600	¼	⅜	⅜	⅜	½	⅝	⅝	¾	⅞	1	1⅛	1¼	1¼	1⅜	1½	1⅝	1¾	1⅞	1⅞	2	2⅛	2¼	2⅜	2½	2½
610	¼	⅜	⅜	⅜	½	⅝	¾	¾	⅞	1	1⅛	1¼	1⅜	1⅜	1½	1⅝	1¾	1⅞	1⅞	2⅛	2⅛	2¼	2⅜	2½	2⅝
620	¼	⅜	⅜	⅜	½	⅝	¾	⅞	⅞	1	1⅛	1¼	1⅜	1½	1⅝	1⅝	1¾	1⅞	2	2⅛	2¼	2⅜	2⅜	2½	2⅝
630	¼	⅜	⅜	⅜	½	⅝	¾	⅞	⅞	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1¾	1⅞	2	2⅛	2¼	2⅜	2½	2⅝	2⅝
640	¼	⅜	⅜	⅜	½	⅝	¾	⅞	1	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1⅞	2	2	2⅛	2¼	2⅜	2½	2⅝	2¾
650	¼	⅜	⅜	½	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1⅜	1½	1⅝	1¾	1⅞	2	2⅛	2¼	2¼	2⅜	2½	2⅝	2¾

660	1/4	3/4	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 3/4
670	1/4	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8
680	1/4	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8
690	1/4	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 1/4	2 1/2	2 5/8	2 3/4	2 3/4	2 7/8
700	1/4	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3
710	1/4	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3
720	1/4	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 3/8	2 1/2	2 1/2	2 5/8	2 3/4	2 7/8	3
730	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/8
740	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/8
750	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/8
760	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 5/8	2 3/4	2 7/8	3	3 1/8	3 1/4
770	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/8	3 1/4
780	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/8	3 1/4
790	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	3	3 1/8	3 1/4	3 3/8
800	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 1/4	2 3/8	2 1/2	2 3/4	2 7/8	3	3 1/8	3 1/4	3 3/8
810	3/8	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/8	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/8	3 1/4	3 3/8
820	3/8	3/8	1/2	5/8	5/8	3/4	7/8	1	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/4	3 3/8	3 1/2
830	3/8	3/8	1/2	5/8	5/8	3/4	7/8	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	1 7/8	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	3	3 1/8	3 1/4	3 3/8	3 1/2
840	3/8	3/8	1/2	5/8	5/8	3/4	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 3/4	2	2 1/8	2 1/4	2 3/8	2 1/2	2 5/8	2 7/8	3	3 1/8	3 1/4	3 3/8	3 1/2
850	3/8	3/8	1/2	5/8	5/8	3/4	1	1 1/8	1 1/4	1 3/8	1 1/2	1 5/8	1 7/8	2	2 1/8	2 1/4	2 1/2	2 5/8	2 5/8	2 7/8	3	3 1/8	3 1/4	3 1/2	3 5/8
860	3/8	3/8	1/2	5/8	5/8	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 3/4	1 7/8	2	2 1/8	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/8	3 3/8	3 1/2	3 5/8
870	3/8	1/2	1/2	5/8	5/8	7/8	1	1 1/8	1 1/4	1 3/8	1 5/8	1 3/4	1 7/8	2	2 1/8	2 3/8	2 1/2	2 5/8	2 3/4	2 7/8	3 1/8	3 1/4	3 3/8	3 1/2	3 5/8
880	3/8	1/2	1/2	5/8	5/8	7/8	1	1 1/8	1 1/4	1 3/8	1 5/8	1 3/4	1 7/8	2	2 1/4	2 3/8	2 1/2	2 5/8	2 3/4	3	3 1/8	3 1/4	3 3/8	3 5/8	3 3/4
890	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 1/2	1 5/8	1 3/4	1 7/8	2 1/8	2 1/4	2 3/8	2 5/8	2 5/8	2 7/8	3	3 1/8	3 1/4	3 1/2	3 5/8	3 3/4
900	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 1/2	1 5/8	1 3/4	1 7/8	2 1/8	2 1/4	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/8	3 3/8	3 1/2	3 5/8	3 3/4
910	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 1/2	1 5/8	1 3/4	2	2 1/8	2 1/4	2 1/2	2 5/8	2 3/4	2 7/8	3	3 1/4	3 3/8	3 1/2	3 3/4	3 7/8
920	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/8	1 3/8	1 1/2	1 5/8	1 7/8	2	2 1/8	2 1/4	2 1/2	2 5/8	2 3/4	2 7/8	3 1/8	3 1/4	3 1/2	3 5/8	3 3/4	3 7/8
930	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/4	1 3/8	1 1/2	1 3/4	1 7/8	2	2 1/8	2 3/8	2 1/2	2 5/8	2 3/4	3	3 1/8	3 1/4	3 1/2	3 5/8	3 3/4	3 7/8
940	3/8	1/2	1/2	5/8	3/4	7/8	1	1 1/4	1 3/8	1 1/2	1 3/4	1 7/8	2	2 1/8	2 3/8	2 1/2	2 3/4	2 3/4	3	3 1/8	3 3/8	3 1/2	3 5/8	3 3/4	4
950	3/8	1/2	1/2	5/8	3/4	7/8	1 1/8	1 1/4	1 3/8	1 1/2	1 3/4	1 7/8	2	2 1/4	2 3/8	2 1/2	2 3/4	2 7/8	3	3 1/4	3 3/8	3 1/2	3 5/8	3 7/8	4
960	3/8	1/2	5/8	5/8	3/4	7/8	1 1/8	1 1/4	1 3/8	1 5/8	1 3/4	1 7/8	2	2 1/4	2 3/8	2 1/2	2 3/4	2 7/8	3	3 1/4	3 3/8	3 5/8	3 3/4	3 7/8	4
970	3/8	1/2	5/8	5/8	3/4	7/8	1 1/8	1 1/4	1 3/8	1 5/8	1 3/4	1 7/8	2 1/8	2 1/4	2 1/2	2 5/8	2 3/4	2 7/8	3 1/8	3 1/4	3 3/8	3 5/8	3 3/4	3 7/8	4 1/8
980	3/8	1/2	5/8	5/8	3/4	7/8	1 1/8	1 1/4	1 3/8	1 5/8	1 3/4	1 7/8	2 1/8	2 1/4	2 1/2	2 5/8	2 3/4	2 7/8	3 1/8	3 1/4	3 1/2	3 5/8	3 3/4	3 7/8	4 1/8
990	3/8	1/2	5/8	5/8	3/4	7/8	1 1/8	1 1/4	1 3/8	1 5/8	1 3/4	2	2 1/8	2 1/4	2 1/2	2 5/8	2 3/4	3	3 1/8	3 1/4	3 1/2	3 5/8	3 3/4	3 7/8	4 1/8
1000	3/8	1/2	5/8	5/8	3/4	7/8	1 1/8	1 1/4	1 1/2	1 5/8	1 3/4	2	2 1/8	2 1/4	2 1/2	2 5/8	2 7/8	3	3 1/4	3 3/8	3 1/2	3 3/4	3 3/4	4	4 1/8

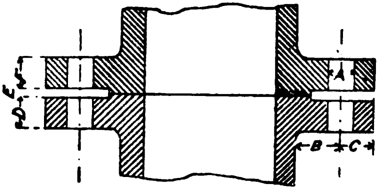


FIG. 3.—FLANGES FOR CAST-IRON PIPES.

TABLE 12.—PROPORTION OF CAST-IRON FLANGES ON CAST-IRON PIPES. (*Fig.* 3.)

A inches.	B inches.	C inches.	D inches.	E inches.	F inches.
½	1	⅞	¾	¼	¾
⅝	1⅛	⅞	⅞	¼	⅞
¾	1¼	1	1	¼	1
⅞	1⅜	1⅛	1⅛	¼	1⅛
1	1½	1¼	1¼	¼	1¼
1⅛	1⅝	1⅜	1⅜	¼	1⅜
1¼	1¾	1½	1⅝	⅜	1⅝
1½	2	1¾	1⅞	½	1⅞
1¾	2¼	2	2¼	½	2¼
2	2½	2¼	2½	½	2½

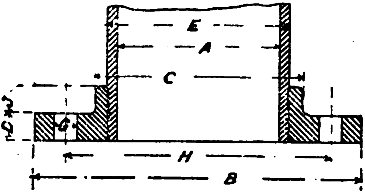


FIG. 4.—FLANGES FOR WROUGHT-IRON PIPES.

TABLE 13.—PROPORTIONS OF FLANGES FOR WROUGHT-IRON PIPES. (*Fig.* 4, page 332.)

A in.	B in.	C in.	D For wrought iron. inches.	D For cast-iron. ins.	E ins.	F Thickness of cast-iron pipe. inches.	G Diameter of bolt. inches.	Number of bolts.	H Pitch of bolts. inches.	J in.
1	4¾	2	½	9/16	1 5/16	½	½	3	3½	½
1¼	5	2¼	½	9/16	1⅝	½	½	3	3¾	½
1½	5¼	2½	½	⅝	1 7/16	½	½	4	4	½
1¾	6	2¾	½	⅝	2 3/32	½	⅝	4	4½	½
2	6¼	3	9/16	11/16	2 5/16	½	⅝	4	4¾	9/16
2¼	6¾	3¼	9/16	11/16	2⅝	½	⅝	4	5⅛	9/16
2½	7	3⅝	9/16	11/16	2 15/16	½	⅝	4	5⅜	9/16
2¾	7¼	3⅞	9/16	11/16	3 3/16	½	⅝	4	5⅝	9/16
3	7¾	4¼	⅝	¾	3 7/16	½	⅝	4	6	⅝
3¼	8	4½	⅝	¾	3¾	9/16	⅝	4	6¼	⅝
3½	8½	4¾	⅝	¾	4	9/16	¾	4	6¾	⅝
3¾	8¾	5	⅝	¾	4¼	9/16	¾	5	7	⅝
4	9	5¼	¾	⅞	4½	9/16	¾	5	7¼	¾
4¼	9¼	5½	¾	⅞	4¾	9/16	¾	5	7½	¾
4½	9½	5¾	¾	⅞	5	9/16	¾	6	7¾	¾
5	10	6¼	¾	1	5½	⅝	¾	6	8¼	¾
5½	10½	6¾	¾	1	6	⅝	¾	6	8¾	¾
6	11	7¼	¾	1	6½	⅝	¾	8	9¼	¾
6½	11½	7¾	¾	1	7	⅝	¾	8	9¾	¾
7	12¼	8¼	¾	1⅛	7½	⅝	¾	8	10¼	¾
7½	12½	8¾	¾	1⅛	8	⅝	¾	8	10¾	¾
8	13	9¼	¾	1⅛	8½	⅝	¾	8	11¼	¾
8½	13½	9¾	¾	1⅛	9	⅝	¾	8	11¾	¾
9	14	10¼	¾	1⅛	9½	⅝	¾	8	12¼	¾
9½	14½	10¾	¾	1⅛	10	⅝	¾	8	12¾	¾
10	15½	11¼	⅞	1¼	10½	⅝	⅞	8	13½	⅞
10½	16	11¾	⅞	1¼	11	⅝	⅞	8	14	⅞
11	16½	12¼	⅞	1¼	11½	⅝	⅞	8	14½	⅞
11½	17¼	13	⅞	1¼	12	⅝	⅞	8	15¼	⅞
12	18½	13⅝	1	1⅜	12½	⅝	1	10	16⅛	1

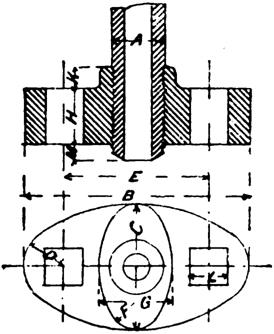


FIG. 5.—OVAL FLANGES.

TABLE 14.—PROPORTIONS OF HEAVY OVAL FLANGES (*Cast Iron*). (*Fig.* 5.)

Bore. in.	A in.	B in.	C in.	D in.	E in.	F in.	G in.	H in.	K in.	L in.	M in.	Size of bolt. inches.
$\frac{1}{2}$	$1\frac{3}{16}$	$3\frac{1}{2}$	$1\frac{3}{4}$	$\frac{5}{8}$	$2\frac{1}{4}$	$\frac{7}{16}$	$1\frac{1}{8}$	$\frac{7}{8}$	$\frac{1}{4}$	$\frac{9}{16}$	$\frac{1}{8}$	$\frac{1}{2}$
$\frac{5}{8}$	$1\frac{1}{16}$	5	$2\frac{1}{2}$	$\frac{13}{16}$	$3\frac{1}{2}$	$\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{8}$	$\frac{11}{16}$	$\frac{1}{8}$	$\frac{5}{8}$
$\frac{3}{4}$	$1\frac{5}{16}$	$5\frac{1}{2}$	$2\frac{3}{4}$	$\frac{7}{8}$	$3\frac{3}{4}$	$\frac{3}{4}$	2	$1\frac{3}{8}$	$\frac{7}{16}$	$\frac{11}{16}$	$\frac{3}{16}$	$\frac{5}{8}$
1	$1\frac{9}{16}$	$6\frac{1}{4}$	3	$\frac{7}{8}$	$4\frac{1}{8}$	$\frac{7}{8}$	$2\frac{1}{4}$	$1\frac{1}{2}$	$\frac{5}{8}$	$\frac{13}{16}$	$\frac{3}{16}$	$\frac{3}{4}$
$1\frac{1}{4}$	$1\frac{7}{8}$	$6\frac{1}{2}$	$3\frac{1}{2}$	1	$4\frac{1}{2}$	1	$2\frac{5}{8}$	$1\frac{5}{8}$	$\frac{5}{8}$	$\frac{13}{16}$	$\frac{1}{4}$	$\frac{3}{4}$
$1\frac{1}{2}$	$2\frac{1}{16}$	$7\frac{1}{4}$	4	$1\frac{1}{8}$	$5\frac{1}{8}$	$1\frac{1}{8}$	3	$1\frac{3}{4}$	$\frac{3}{4}$	$\frac{15}{16}$	$\frac{1}{4}$	$\frac{7}{8}$
$1\frac{3}{4}$	$2\frac{5}{16}$	$8\frac{1}{4}$	$4\frac{1}{2}$	$1\frac{1}{4}$	$5\frac{3}{4}$	$1\frac{5}{16}$	$3\frac{1}{2}$	$1\frac{7}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	$\frac{5}{16}$	$\frac{7}{8}$
2	$2\frac{9}{16}$	9	5	$1\frac{13}{16}$	$6\frac{1}{2}$	$1\frac{3}{4}$	4	2	1	$1\frac{1}{16}$	$\frac{3}{8}$	1

TABLE 15.—PROPORTIONS OF LIGHT OVAL FLANGES (*Cast Iron*). (*Fig.* 5.)

Bore. in.	A in.	B in.	C in.	D in.	E in.	F in.	G in.	H in.	K in.	L in.	M in.	Size of bolt. inches.
1	$1\frac{5}{16}$	5	$2\frac{1}{2}$	$\frac{13}{16}$	$3\frac{1}{2}$	$\frac{5}{8}$	$1\frac{3}{4}$	$\frac{3}{4}$	$\frac{1}{4}$	$\frac{11}{16}$	$\frac{3}{16}$	$\frac{5}{8}$
$1\frac{1}{4}$	$1\frac{9}{16}$	$5\frac{1}{2}$	$2\frac{3}{4}$	$\frac{7}{8}$	$3\frac{3}{4}$	$\frac{3}{4}$	2	$\frac{7}{8}$	$\frac{1}{4}$	$\frac{11}{16}$	$\frac{1}{4}$	$\frac{5}{8}$
$1\frac{1}{2}$	$1\frac{15}{16}$	$6\frac{1}{4}$	3	$\frac{7}{8}$	$4\frac{1}{8}$	$\frac{7}{8}$	$2\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{13}{16}$	$\frac{1}{4}$	$\frac{3}{4}$
$1\frac{3}{4}$	$2\frac{1}{16}$	$6\frac{1}{2}$	$3\frac{1}{2}$	1	$4\frac{1}{2}$	1	$2\frac{5}{8}$	$1\frac{1}{8}$	$\frac{3}{8}$	$\frac{13}{16}$	$\frac{1}{4}$	$\frac{3}{4}$
2	$2\frac{5}{16}$	$7\frac{1}{4}$	4	$1\frac{1}{8}$	$5\frac{1}{8}$	$1\frac{1}{8}$	3	$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{15}{16}$	$\frac{1}{4}$	$\frac{7}{8}$
$2\frac{1}{2}$	$2\frac{15}{16}$	$8\frac{1}{4}$	$4\frac{1}{2}$	$1\frac{1}{4}$	$5\frac{3}{4}$	$1\frac{5}{16}$	$3\frac{1}{2}$	$1\frac{3}{8}$	$\frac{9}{16}$	$\frac{15}{16}$	$\frac{1}{4}$	$\frac{7}{8}$

TABLE 16.

Amount of water required to develop a given horse-power, with a given available effective head.

Effective Head in feet.	Horse-power based on 85%, efficiency of the water wheel.									
	10	20	30	40	50	60	70	80	90	100
	Flow in cu. ft. of water per min. required to develop power.									
50	125	250	375	500	625	750	875	1000	1125	1250
60	104	208	312	416	520	624	726	830	934	1038
70	88	177	266	355	444	532	621	709	798	886
80	77	155	232	311	388	466	544	622	699	896
90	70	140	210	280	350	420	490	560	630	700
100	63	125	186	248	312	372	435	498	558	622
110	59	118	176	234	293	350	410	467	525	585
120	52	104	156	208	260	312	364	415	467	520
130	48	96	143	192	240	287	335	385	430	478
140	45	89	133	178	222	266	310	355	400	443
150	42	83	125	166	208	250	292	332	375	416
160	39	78	117	155	195	233	272	312	350	388
170	37	73	110	146	183	220	256	293	330	365
180	35	69	104	138	172	207	242	276	310	345
190	33	65	98	132	164	198	230	262	295	326
200	31	62	93	124	155	186	218	248	280	310
210	30	59	89	118	148	177	206	236	266	295
220	28	57	85	113	141	159	198	225	255	283
230	27	54	81	108	135	162	190	216	243	270
240	26	52	78	104	130	155	181	207	233	258
250	25	50	75	100	125	149	174	199	224	248
260	24	48	72	96	120	144	167	191	215	238
270	23	46	69	92	115	138	161	184	207	230
280	22	45	67	89	111	133	156	178	200	222
290	21	43	65	86	107	129	150	172	193	215
300	20	42	62	83	104	124	145	166	187	208
310	19	41	60	80	100	120	140	160	180	200
320	19	40	59	78	97	117	136	156	175	194
330	19	38	57	76	94	113	132	151	170	188
340	18	37	55	74	92	110	125	146	165	183
350	18	36	53	71	89	106	124	142	160	178
360	18	35	52	69	86	102	121	138	155	172
370	17	34	50	67	84	100	117	134	151	162
380	17	33	49	66	82	98	114	130	147	164
390	16	32	48	64	80	96	111	127	144	160
400	16	31	47	63	77	94	105	124	140	156

TABLE 17.—CONTENTS OF CIRCULAR TANKS IN ENGLISH GALLONS.

Depth of tank in feet.	Diameter of tank in feet.																
	5	6	7	8	9	10	11	12	13	14	15	16	18	20	22	24	25
	Capacity of tank in English gallons.																
5	610	880	1199	1566	1982	2447	2961	3524	4136	4797	5506	6265	7929	9790	11875	14097	15296
6	734	1057	1439	1879	2378	2937	3553	4229	4963	5756	6608	7518	9515	11748	14251	16917	18356
7	856	1233	1689	2192	2775	3426	4146	4934	5790	6715	7709	8771	11101	13706	16626	19736	21415
8	979	1469	1918	2406	3171	3916	4778	5639	6618	7675	8811	10024	12687	15664	19001	22556	24475
9	1100	1585	2158	2819	3568	4405	5330	6343	7445	8634	9912	11278	14273	17622	21376	25375	27534
10	1220	1762	2398	3132	3964	4895	5922	7048	8272	9594	11013	12531	15859	19580	23751	28195	30593
11	1346	1938	2678	3446	4361	5384	6515	7753	9099	10553	12115	13784	17444	21538	26126	31014	33653
12	1468	2114	2878	3759	4757	5874	7107	8458	9927	11513	13216	15037	19031	23496	28502	33834	36712
13	1590	2290	3116	4072	5154	6363	7699	9163	10754	12472	14317	16290	20617	25454	30877	36653	39773
14	1713	2467	3357	4385	5550	6853	8292	9868	11581	13431	15419	17543	22203	27412	33252	39473	42831
15	1835	2643	3597	4699	5937	7342	8884	10573	12408	14391	16520	18796	23789	29370	35627	42292	45890
16	1958	2819	3837	5012	6343	7832	9476	11278	13236	15350	17622	20049	25375	31328	38002	45112	48950
17	2080	2995	4077	5325	6740	8321	10069	11982	14063	16310	18723	21303	26961	33286	40378	47931	52009
18	2202	3171	4317	5539	7136	8811	10661	12687	14890	17269	19824	22556	28547	35244	42753	50751	55068
19	2324	3348	4557	5952	7533	9300	11253	13392	15717	18228	20926	23809	30133	37202	45128	53571	58128
20	2447	3524	4797	6265	7929	9790	11845	14097	16545	19188	22027	25062	31719	39160	47503	56390	61187
21	2569	3700	5036	6578	8326	10279	12438	14802	17372	20147	23128	26315	33305	41118	49878	59209	64246
22	2692	3876	5276	6892	8722	10769	13020	15507	18199	21107	24230	27568	34891	43076	52253	62029	67306
23	2814	4053	5516	7205	9119	11258	13622	16222	19026	22066	25331	28821	36477	45034	54629	64848	70365
24	2937	4229	5756	7518	9515	11748	14214	16917	19854	23026	26433	30074	38063	46992	57004	67668	73425
25	3059	4405	5996	7832	9912	12237	14807	17622	20681	23985	27534	31328	39649	48950	59379	70488	76484
26	3181	4580	6236	8145	10308	12727	15399	18326	21508	24944	28635	32581	41335	50908	63754	73297	79543

TABLE 18.—CONTENTS OF RECTANGULAR TANKS, IN ENGLISH GALLONS.

Gallons about	Outside dimensions.						Gallons about	Outside dimensions.					
	Lengths		Widths		Depths			Lengths		Widths		Depths	
	ft.	in.	ft.	in.	ft.	in.		ft.	in.	ft.	in.	ft.	in.
20	1	8	1	3	1	6	50	2	6	1	11	1	10
20	2	0	1	0	1	8	60	3	1	2	1	1	9
20	2	0	1	8	1	0	70	3	1	2	1	2	0
20	3	2	1	0	1	0	80	3	1	2	3	2	1
25	2	1	1	9	1	3	100	3	1	2	7	2	3
25	2	1	1	1	2	0	125	3	3	2	7	2	6
25	3	3	1	4	1	0	150	3	8	2	11	2	6
30	2	1	2	1	1	3	175	3	11	3	1	2	7
30	2	1	1	4	2	0	200	4	1	3	1	2	9
30	3	3	1	7	1	0	250	5	1	3	1	2	9
30	2	1	1	7	1	7	300	6	1	3	1	2	9
40	2	1	2	1	1	7	350	6	6	3	3	2	9
40	2	1	1	8	2	0	400	7	1	3	7	2	9
40	3	4	2	1	1	0	500	8	1	3	10	2	9
40	2	4	1	9	1	8	600	8	1	4	1	3	1
50	2	8	2	1	1	7	800	8	1	4	7	3	8
50	2	1	2	1	2	0	1000	9	1	5	1	8	8
50	3	4	2	7	1	0							

TABLE 19.—APPROXIMATE WEIGHT OF WROUGHT-IRON TANKS, *plates ¼ inch thick.*

Capacity 100 gallons = Weight 3 cwts. 3 qrs. 4 lbs.
,, 150 ,, = ,, 5 ,, 0 ,, 0 ,,
,, 200 ,, = ,, 7 ,, 0 ,, 0 ,,
,, 250 ,, = ,, 8 ,, 0 ,, 0 ,,
,, 300 ,, = ,, 9 ,, 0 ,, 0 ,,
,, 350 ,, = ,, 10 ,, 0 ,, 0 ,,
,, 400 ,, = ,, 11 ,, 0 ,, 0 ,,
,, 450 ,, = ,, 12 ,, 0 ,, 0 ,,
,, 500 ,, = ,, 13 ,, 0 ,, 0 ,,
,, 1000 ,, = ,, 26 ,, 0 ,, 0 ,,
,, 1500 ,, = ,, 30 ,, 0 ,, 0 ,,
,, 2000 ,, = ,, 47 ,, 0 ,, 0 ,,
,, 3000 ,, = ,, 72 ,, 0 ,, 0 ,,
,, 4000 ,, = ,, 80 ,, 0 ,, 0 ,,

AIR.

MEMORANDA ON AIR.

Pressure of one atmosphere = 14·7 lbs. per square inch.
= 2116·4 lbs. per square foot.
= 1·0335 kilogrammes per square centimetre.
= 29·922 inches of mercury at 32 degrees Fahr.
= 76 centimetres of mercury at 32 degrees Fahr.
= 30 inches of mercury at 62 degrees Fahr.
= 33·947 feet of water at 62 degrees Fahr.
= 10·347 metres of water at 62 degrees Fahr.

One cubic foot, at 14·7 lbs. per square inch, or one atmosphere = 0·080728 lb. at 32 degrees Fahr.
= 1·29 ounces at 32 degrees Fahr.
= 565·1 grains at 32 degrees Fahr.

One cubic foot, at 14·7 lbs. per square inch, or one atmosphere = 0·076097 lb. at 62 degrees Fahr.
= 1·217 ounces at 62 degrees Fahr.
= 532·7 grains at 62 degrees Fahr.

1 pound per square inch = 2·035 inches of mercury at 32 degrees Fahr.
= 51·7 millimetres at 32 degrees Fahr.
= 2·04 inches of mercury at 62 degrees Fahr.
= 2·31 feet of water at 62 degrees Fahr.
= 27·72 inches of water at 62 degrees Fahr.

1 ounce per square inch = 1·732 inches of water at 62 degrees Fahr.

1 ounce per square foot = 0·1925 inch of water at 62 degrees Fahr.
= 0·01417 inch of mercury at 62 degrees Fahr.

1 litre, under one atmosphere pressure = 1·293 grammes at 32 degrees Fahr.
= 19·955 grains at 32 degrees Fahr.

1 pound of air at 62 degrees Fahr. = 13·141 cubic feet.

To find the volume of one pound of air under 14·7 lbs. pressure per square inch at a given temperature.

$$V = \frac{T + 461}{39 \cdot 80};$$

When V = volume of air in cubic feet;
T = temperature in degrees Fahr.

To find the volume of one pound of air of a given temperature and pressure.

$$V = \frac{T + 461}{2 \cdot 7074 \times P};$$

When P = pressure in lbs. per square inch.

To find the pressure in lbs. per square inch of one lb. of air at a given temperature and volume.

$$P = \frac{T + 461}{2 \cdot 7074 \times V};$$

To find the temperature of one lb. of air of a given volume and pressure.

$$T = 2 \cdot 7074 \times V \times P - 461.$$

To find the density of air at a given temperature and volume.

$$D = 2 \cdot 7074 \frac{P}{T + 461};$$

When D = density of one cubic foot of air at atmospheric pressure in lbs.

TABLE 20.—THE CO-EFFICIENT 2.7074 FOR OTHER GASES ARE GIVEN IN FOLLOWING TABLE.

Name of Gas.	Volume of one lb. of gas at 32° Fahr. under one atmosphere of pressure.	Value of Co-efficient.	
	Cubic feet.	A	A_1
Air	12·387	0·36935	$\frac{1}{2{\cdot}7074}$
Carbonic acid (ideal)	8·157	0·24322	$\frac{1}{4{\cdot}1114}$
Carbonic acid (actual)	8·101	0·24155	$\frac{1}{4{\cdot}1399}$
Ether vapour .	4·777	0·14246	$\frac{1}{7{\cdot}0195}$
Gaseous steam . .	19·913	0·59372	$\frac{1}{1{\cdot}6842}$
Hydrogen . .	178·830	5·33200	$\frac{1}{0{\cdot}1875}$
Nitrogen . . .	12·753	0·38027	$\frac{1}{2{\cdot}6297}$
Olefiant gas . .	12·580	0·37506	$\frac{1}{2{\cdot}6662}$
Oxygen . . .	11·205	0·33406	$\frac{1}{2{\cdot}9935}$
Vapour of mercury .	1·776	0·05296	$\frac{1}{18{\cdot}878}$

TABLE 21.—DENSITY, VOLUME AND PRESSURE OF AIR AT VARIOUS TEMPERATURES (D. K. Clark).

Temperature Fahr. Scale.	Volume of one lb. of air at constant atmospheric pressure, 14·7 lbs. per square inch. Datum = volume at 62° Fahr. = 1.		Density of weight of one cubic foot of air at atmospheric pressure.	Pressure at a given weight of air having a constant volume. Datum = atmospheric pressure at 62° Fahr. = 1.	
Degrees.	Cubic feet.	Comparative volume.	lbs. per square inch.	lbs. per square inch.	Comparative pressure.
0	11·583	0·881	0·086331	12·96	0·881
32	12·387	0·943	0·080728	13·86	0·943
40	12·586	0·958	0·079439	14·08	0·958
50	12·840	0·977	0·077884	14·36	0·977
62	13·141	1·000	0·076097	14·70	1·000
70	13·342	1·015	0·074950	14·92	1·015
80	13·593	1·034	0·073565	15·21	1·034
90	13·845	1·054	0·072230	15·49	1·054
100	14·096	1·073	0·070942	15·77	1·073
120	14·592	1·111	0·068500	16·33	1·111
140	15·100	1·149	0·066221	16·89	1·149
160	15·603	1·187	0·064088	17·50	1·187
180	16·106	1·226	0·062090	18·02	1·226
200	16·606	1·264	0·060210	18·58	1·264
210	16·860	1·283	0·059313	18·86	1·283
212	16·910	1·287	0·059135	18·92	1·287
220	17·111	1·302	0·058442	19·14	1·302
230	17·362	1·321	0·057596	19·42	1·321
240	17·612	1·340	0·056774	19·70	1·340
250	17·865	1·359	0·055975	19·98	1·359
260	18·116	1·379	0·055200	20·27	1·379
270	18·367	1·398	0·054444	20·55	1·398
280	18·621	1·417	0·053710	20·83	1·417
290	18·870	1·436	0·052994	21·11	1·436
300	19·121	1·455	0·052297	21·39	1·455
320	19·624	1·493	0·050959	21·95	1·493
340	20·126	1·532	0·049686	22·51	1·532
360	20·630	1·570	0·048476	23·08	1·570
380	21·131	1·608	0·047323	23·64	1·608
400	21·634	1·646	0·046223	24·20	1·646

TABLE 21 (*Continued*).

Temperature Fahr. Scale.	Volume of one lb. of air at constant atmospheric pressure, 14·7 lbs. per square inch. Datum = volume at 62° Fahr. = 1.		Density of weight of one cubic foot of air at atmospheric pressure.	Pressure at a given weight of air having a constant volume. Datum = atmospheric pressure at 62° Fahr. = 1.	
Degrees.	Cubic feet.	Comparative volume.	lbs. per square inch.	lbs. per square inch.	Comparative pressure.
425	22·262	1·694	0·044920	24·90	1·694
450	22·890	1·742	0·043686	25·61	1·742
475	23·518	1·789	0·042520	26·31	1·789
500	24·146	1·837	0·041414	27·01	1·837
525	24·775	1·885	0·040364	27·71	1·885
550	25·403	1·933	0·039365	28·42	1·933
575	26·031	1·981	0·038415	29·12	1·981
600	26·659	2·029	0·037510	29·82	2·029
650	27·915	2·124	0·035822	31·23	2·124
700	29·172	2·220	0·034280	32·63	2·220
750	30·428	2·315	0·032865	34·04	2·315
800	31·685	2·411	0·031561	35·44	2·411
850	32·941	2·507	0·030358	36·85	2·507
900	34·197	2·602	0·029242	38·25	2·602
950	35·453	2·698	0·028206	39·66	2·698
1000	36·710	2·793	0·027241	41·06	2·793
1500	49·274	3·749	0·020295	55·12	3·749
2000	61·836	4·705	0·016172	69·17	4·705
2500	74·400	5·661	0·013441	83·22	5·661
3000	86·962	6·618	0·011499	97·28	6·618

AIR FLOW FORMULAE.

To find the velocity of air in pipes.

$$V = 396 \sqrt{\frac{H \times D}{L}};$$

When V = velocity of air in feet per second;
H = head or friction, in inches of water;
D = diameter of pipe in feet;
L = length of pipe in feet;

To find head or drag of air.

$$H = \frac{L \times V^2}{156{,}800 \times D}.$$

To find the horse-power required for net work of discharging air.

$$HP = \frac{V \times D^2 \times H}{135}$$

$$HP = \frac{V^3 \times D^2 \times L}{21{,}200{,}000};$$

When HP = effective horse-power required for net work of discharging air.

To find the head due to friction of air.

$$H = \frac{C^2 \times L}{(3{\cdot}7\ D)^5};$$

When H = the head, or difference of pressure at the two ends of the pipe in inches of water;
C = cubic feet of air per minute;
L = the length of pipe in yards;
D = the diameter of pipe in inches.

To find cubic feet per minute flowing through a pipe.

$$C = \sqrt{\frac{(3{\cdot}7\ D)^5 \times H}{L}}$$

$$(3{\cdot}7\ D)^5 = \frac{C^2 \times L}{H};$$

$$L = \frac{(3{\cdot}7\ D)^5}{C^2} \times H;$$

E 22.—FLOW OF AIR THROUGH AN ORIFICE.

lume and Horse-power required when air under given ozs. per sq. in. is allowed to escape into the atmosphere.

Pressure inches f water.	Velocity feet per min.	Volume through 1 sq. in. area c. ft. per min.	Horse-power to move the given volume of air.	Horse-power per 1,000 c. ft. per min.
0·216	1·828	12·69	0·00043	0·0340
0·432	2·585	17·95	0·00122	0·0680
0·648	3·165	21·98	0·00225	0·1022
0·864	3·654	25·37	0·00346	0·1363
1·080	4·084	28·36	0·00483	0·1703
1·296	4·473	31·06	0·00635	0·2044
1·512	4·830	33·54	0·00800	0·2385
1·728	5·162	35·85	0·00978	0·2728
1·944	5·473	38·01	0·01166	0·3068
2·160	5·768	40·06	0·01366	0·3410
2·376	6·048	42·00	0·01575	0·3750
2·592	6·315	43·86	0·01794	0·4090
2·808	6·571	45·63	0·02022	0·4431
3·024	6·818	47·34	0·02260	0·4772
3·240	7·055	49·00	0·02505	0·5112
	7·284	50·59	0·02759	0·5454
	7·507	52·13	0·03021	0·5795
	7·722	53·63	0·03291	0·6136
	7·932	55·08	0·03568	0·6476
	8·136	56·50	0·03852	0·6818
	8·334	57·88	0·04144	0·7160
	8·528	59·22	0·04442	0·7500
	8·718	60·54	0·04749	0·7841
	8·903	61·83	0·05058	0·8180
	9·084	63·08	0·05376	0·8522
	9·262	64·32	0·05701	0·8863
	9·435	65·52	0·06031	0·9205
	9·606	66·71	0·06368	0·9546
	9·773	67·87	0·06710	0·[illegible]887
	9·938	69·01	0·07058	1·0227
	10·100	70·14	0·07412	1·0567

TABLE 23.—FRICTION OF AIR AND GAS IN LONG PIPES.

Diameter of pipe in inches.	Head, or difference of pressure at the two ends of a pipe one yard long.	
	Inches of water.	Pounds per square inch.
	Head for 1 cubic foot per minute.	
¼	1·477	0·05317
⅜	0·1945	0·00700
½	0·0185	0·000666
¾	0·006077	0·0002187
1	0·001442	0·0000519
	Head for 10 cubic feet per minute.	
1¼	0·04725	0·001701
1½	0·01899	0·000684
1¾	0·00878	0·000326
2	0·00451	0·000162
2¼	0·00250	0·000090
2½	0·00148	0·0000532
	Head for 100 cubic feet per minute.	
3	0·0594	0·00214
3½	0·0746	0·000989
4	0·0141	0·000507
4½	0·00782	0·000281
5	0·00462	0·000166
6	0·00186	0·0000668
	Head for 100 cubic feet per minute.	
7	0·000858	0·000309
8	0·000440	0·0000158
	Head for 1000 cubic feet per minute.	
9	0·02442	0·000879
10	0·01442	0·000519
12	0·00579	0·000209
15	0·00189	0·0000681
18	0·000763	0·0000275
21	0·000353	0·0000127
24	0·000181	0·00000653
	Head for 10,000 cubic feet per minute.	
30	0·00593	0·000214
33	0·00368	0·000133
36	0·00238	0·0000858
42	0·001103	0·0000398
48	0·000566	0·0000204
54	0·000314	0·0000113
60	0·000185	0·00000667

In order to facilitate calculation, the above Table is so arranged, that for diameters under one inch the discharge must be taken in cubic feet, but from 1¼ to 2½ inches in tens, from 3 to 8 inches in hundreds, from 9 to 24 inches in thousands, and from 30 to 60 inches in tens of thousands of cubic feet.

$$H = K \times C^2 \times L;$$

$$D = \frac{C}{\sqrt{K} \times \sqrt{H}};$$

When K = co-efficient in above Table;
D = assumed discharge in cubic feet.
H = head of friction.

Table 24.—Volumes and Weights of Gases.

Name of Gases.	Pound in one cubic foot.	Cubic feet in one pound.
Hydrogen	0·00559	178·83
Oxygen	0·0895	11·18
Nitrogen	0·0783	12·77
Carbonic oxide	0·0783	12·77
Carbon dioxide	0·123	8·13
Marsh gas	0·0447	22·35
Ethylene	0·0783	12·77
Benzene	0·2181	4·58
Atmospheric air	0·08073	12·39

Table 25.—Specific Heat of Gases.

Name of Gases.	Specific heat at constant volume.	Specific heat at constant pressure.
Air	0·169	0·238
Oxygen	0·156	0·218
Hydrogen	2·410	3·405
Nitrogen	0·173	0·244
Carbonic oxide	0·173	0·244
Carbon dioxide	0·171	0·216
Marsh gas	0·470	0·593
Ethylene	0·332	0·404
Steam	0·369	0·480

SECTION VI

POWER PLANT

Internal Combustion Engines (*excepting* Diesel Engines)
A. **Garrard**, Wh.Ex., A.F.R.Ac.S., M.Illg.Eng.Soc.

Diesel Engines . A. P. **Chalkey**, B.Sc., A.M.I.C.E.

Steam and Steam Engines (*excepting* Steam Turbines)
R. E. **Neale**, B.Sc., A.M.I.E.E., A.G.C.I.

Steam Turbines . . R. **Dowson** B.Sc., A.M.I.C.E.

Steam Tables . . . . . **The Editor.**

Fuels, Combustion, Steam Generators
D. **Brownlie**, B.Sc., F.C.S., A.M.I.Min.E., A.I.Mech.E.

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INTERNAL COMBUSTION ENGINES.

GENERAL PRINCIPLES.

The internal combustion engine belongs to the class of heat engine in which heat is imparted to and abstracted from a quantity of gaseous fluid contained in the cylinder, this fluid when heated expanding and forcing the piston outwards and when cooled exerting a smaller pressure on the piston so that it can be readily returned by the fly-wheel or otherwise. The net work done per revolution will then be the difference between that done by the expanding gas on the outward stroke and that done by the flywheel or other means by which the piston is returned. This definition, of course, covers hot-air engines, which are *external combustion engines,* and are of very little practical importance. The internal combustion engine is characterised by the method of imparting heat to the working fluid, which is primarily air. In order to heat this air rapidly at the beginning of the working stroke and thus raise its pressure, a small quantity of fuel in the form of a gas, vapour or finely divided liquid is mixed with the air and ignited. In this way a quantity of heat is imparted with extreme rapidity to the

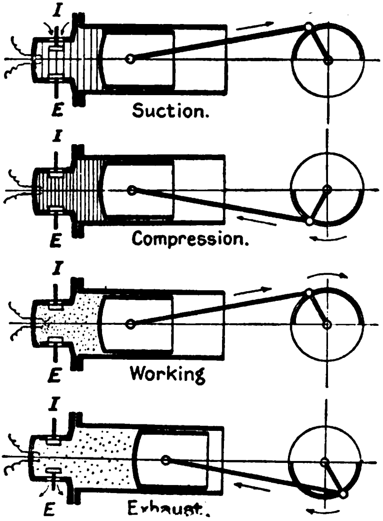


FIG. 1.

working fluid and raises its temperature and pressure so that the piston is forced forward on the working stroke. The nitrogen of the air and the products of the combustion, instead of being cooled like the air in a hot-air engine, are exhausted and a fresh charge of air and combustible gas or vapour is drawn into the cylinder. The only methods now in general use by which (*a*) the cylinder is supplied with a mixture of air and fuel and (*b*) such mixture after combustion is exhausted, are as follows—

Four-stroke Cycle. In this cycle which was first proposed by M. Beau de Rochas, but practically developed by Dr. Otto, one working stroke occurs in every two revolutions, the other three strokes being preparatory ones only. The cycle of operations is shown diagrammatically in Fig. 1. The mixture in the cylinder before combustion is indicated by lines and the products of combustion by dots.

Two-stroke Cycle. This cycle needs a separate pump, which in small engines is generally constituted by the piston and crank chamber. The cycle consists of a compression and a working stroke with intervening exhaust and inlet periods as shown diagrammatically in Fig. 2. The non-return valve A admits the mixture to the crank chamber from which it passes through the transfer port I to the cylinder. The products of combustion are exhausted at one side of the cylinder before the end of the stroke is reached and the fresh charge enters at the other side from the pump, which has previously compressed it to a pressure of 6 to 8 lbs. per square inch.

Ideal Efficiency. Ideal internal combustion engines working on this principle of the reception and rejection of heat necessarily possess an efficiency which is less than unity. In the Carnot ideal cycle, the efficiency is—

$$\frac{T_1 - T_2}{T_1} \quad . \quad . \quad . \quad . \quad (1)$$

where T_1 and T_2 are the maximum and minimum temperatures in the cylinder, otherwise the temperatures at which heat is supplied to and rejected by the working fluid. Engines cannot be made to work practically on the Carnot cycle, one reason being the very low mean effective pressure. Internal combustion engines of to-day, whether working on the four or two-stroke cycles, approximate more or less to one or the other of two ideal types as follows—

Constant-volume Type. Gas engines, petrol engines, paraffin-vapour engines, in fact all engines in which a mixture of air and gas or vapour is compressed are of the constant-volume type. A constant-volume cycle is shown in Fig. 3, which only departs from the ideal constant volume cycle in that the products of combustion expand only to the original volume of the mixture and not to the original pressure (atmospheric). It is assumed in this cycle that heat is instantaneously applied at the moment of maximum pressure when the piston is on its dead centre. The explosion of the mixture of course only approximates to this. The formula for the efficiency of this engine assuming the ideal cycle followed is—

$$\text{Efficiency} = 1 - \left(\frac{1}{r}\right)^{\gamma - 1} \quad . \quad . \quad . \quad . \quad (2)$$

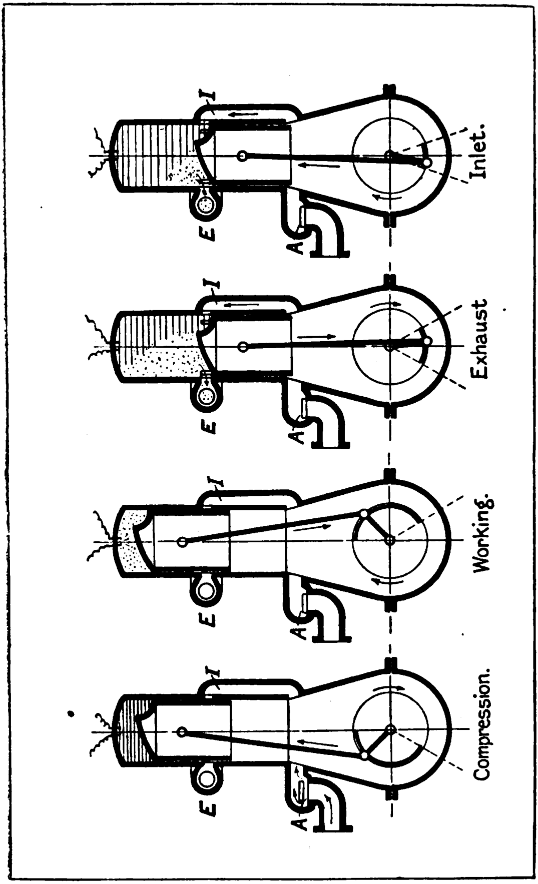


FIG. 2.

where r is the compression ratio, that is the ratio of the volumes of the mixture at the beginning and at the end of the compression stroke, and γ is the ratio of the specific heats of the working fluid; γ may be taken as 1·4. It will be seen that the efficiency depends purely upon the compression ratio. The efficiency formula for the ideal cycle is simple but it becomes more complicated if account is taken of the loss due to the heat rejected by the release of the products before expansion to atmospheric pressure.

In the actual diagram shown, $r = 5$, and the maximum compression pressure is 142 lbs. per square inch. The combustion raises the pressure to 500 and expansion takes place adiabatically until the pressure is 52. The absolute temperatures at various stages of the cycle are shown in the diagram.

The efficiency according to formula (2) is ·48. The mean pressure is 105 lbs. per square inch. Expansion to atmospheric pressure as in the ideal cycle is never attempted in practice. The small gain is more than offset by the mechanical difficulties and losses in the much larger engines required.

The value $r = 7$ is the highest in ordinary use in constant volume engines, while in constant pressure engines (Diesel) the value $r = 12$ to 14 has not so far been exceeded.

Constant-pressure Type. The constant-pressure cycle shown diagrammatically in Fig. 4 corresponds to the constant volume cycle of Fig. 3 as regards maximum pressure. Diesel engines and fuel injection engines fall under this heading, since the fuel is sprayed in during the early part of the working stroke, so that there is practically a continuous burning during this short period, instead of the explosion resulting from ignition of a mixture. After the injection period is completed the hot gas is expanded as before. The efficiency of ideal constant-pressure engines is also given by the formula (2) used above.

In the example shown, $r = 12{\cdot}24$, and the pressure to which the charge of air is compressed is 500 lbs. per square inch. The combustion raises the temperature and increases the volume without affecting the pressure. The mixture expands to a pressure of 52 lbs. per square inch. The mean pressure is 117 lbs. per square inch. The efficiency by the ideal formula (2) is ·64, but if account is taken of the loss due to the heat rejected in the exhaust products at the end of the expansion the efficiency becomes ·56.

Engines attain to an efficiency in practice of about 70 per cent of their ideal efficiency. The difference is due largely (to the extent of about 18 per cent) to the fact that the specific heat of the actual working fluid is not constant but increases with temperature. The remaining 12 per cent is the real margin available for improvement.

Adiabatic and Isothermal Curves. All curves of pressure and volume in ideal internal combustion engines follow a law of the form

$$p \times v^n = c \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where n and c are constants. In actual engines the compression

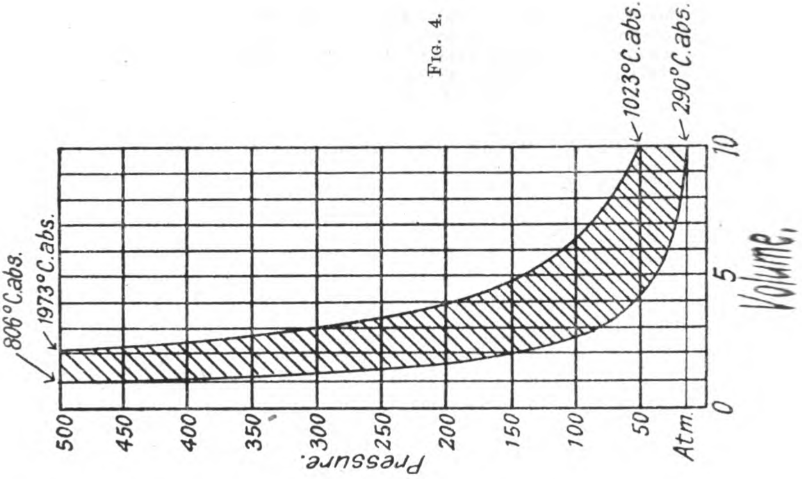

806°C.abs.
1973°C.abs.
1023°C.abs.
290°C.abs.
500
450
400
350
300
250
200
150
100
50
Atm.
Pressure.
0
5
10
Volume.

FIG. 4.

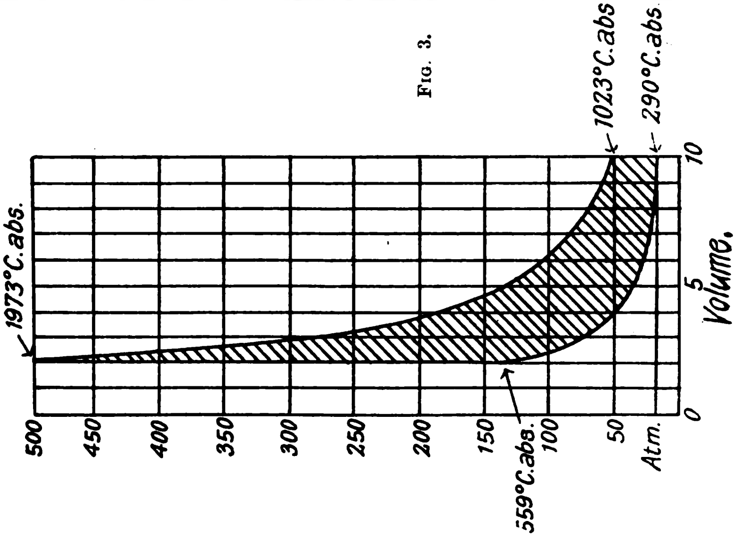

1973°C.abs.
1023°C.abs
290°C.abs
559°C.abs.
500
450
400
350
300
250
200
150
100
50
Atm.
0
5
10
Volume.

FIG. 3.

and expansion curves also follow such a law very closely, but the value of n varies in accordance with conditions. When any gas contracts or expands adiabatically, that is *without giving out or receiving heat energy*, n has a value which is equal to the ratio of the specific heat at constant pressure to the specific heat at constant volume. In the case of air this ratio is 1·4. When the expansion or contraction is isothermal (that is to say, heat is absorbed by the gas during expansion and given out during compression so that *the temperature remains the same*) n has the value unity. In actual heat engines n varies between these limits.

Example. Compression ratio 4. At beginning of adiabatic compression (when $n = 1{\cdot}4$), $p = 15$ lbs. per square inch, and $v = 1$ cubic foot.

Then $pv^n = 15 \times 1^n = 15$. At end of compression stroke, $v = \frac{1}{4}$ and $p. ({\cdot}25)^{1{\cdot}4} = 15$.

Taking logarithms, $\log p + 1{\cdot}4 \log {\cdot}25 = \log 15$,
from which $\log p = 2{\cdot}0189$,
and $p = 104{\cdot}5$ lbs. per square inch.

If the compression had been isothermal instead of adiabatic, the condition at the beginning would be represented by $pv = 15$. When $v = {\cdot}25$, $p = \frac{15}{\cdot 25} = 60$ pounds per square inch.

The above formulae for adiabatic and isothermal changes are only variations of the general formula for a perfect gas, viz.

$$\frac{pv}{T} = \text{constant} \quad . \quad . \quad . \quad (4)$$

T is the absolute temperature, *i.e.* the temperature as measured from the absolute zero of temperature, which is − 273° C. or − 466·7° F., so that absolute temperature is obtained by adding 273 to the temperature in degrees Centigrade or 466·7 to the temperature in degrees Fahrenheit.

Compression Pressures. The actual pressure at the end of compression depends upon r, upon the initial pressure, and upon the value of n in the formula $pv^n = C$. The diagram, Fig. 5, gives the compression pressures for values of r from 3 to 8, and for initial pressure of 13, 14, 15, and 16. The value of n employed in drawing the curves is 1·34.

Work done during Expansion and Compression. The work done on the gas during its compression from $p_1\,v_1$ to $p_2\,v_2$ or *vice versâ* its expansion between these limits, is represented by the shaded area, Fig. 6, and is given by the following formula obtained by integration—

$$w = \frac{p_2\,v_2 - p_1\,v_1}{n - 1} \quad . \quad . \quad . \quad . \quad (5)$$

$$\text{or } w = \frac{p_2\,v_2}{n - 1}\left\{1 - \left(\frac{v_1}{v_2}\right)^{1-n}\right\} \quad . \quad . \quad (6)$$

$$\text{or } w = \frac{p_1\,v_1}{n - 1}\left\{\left(\frac{v_2}{v_1}\right)^{1-n} - 1\right\} \quad . \quad . \quad . \quad (7)$$

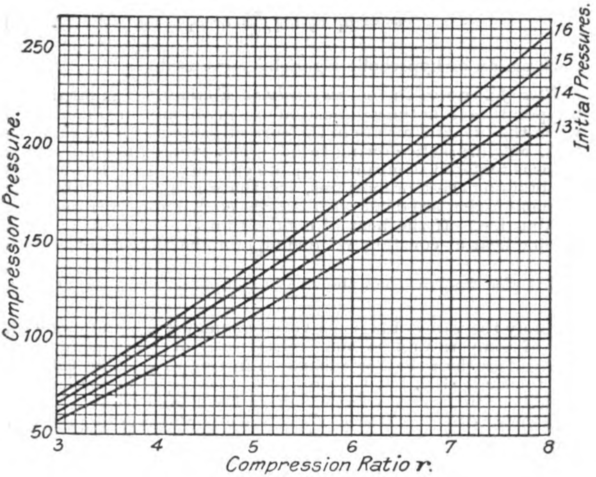

Compression Pressure.
250
200
150
100
50
3
4
5
6
7
8
Compression Ratio r.
Initial Pressures.
16
15
14
13

FIG. 5.

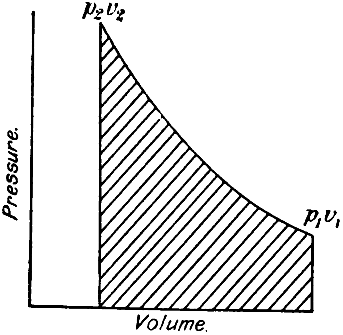

$p_2 v_2$
$p_1 v_1$
Pressure.
Volume.

FIG. 6.

In the special case when $n = 1$ (viz. isothermal variation),

$$w = p_1 v_1 \log_e \frac{v_2}{v_1} \text{ or } p_1 v_1 \log_e \frac{v_1}{v_2} \quad . \quad (7)$$

Naperian or hyperbolic logarithms are used.

Rise of Pressure in Constant-volume Engines. The maximum temperature from the combustion of a given mixture is obtained when the fuel and air are in the correct proportions for combustion, but in practice owing to imperfect mixing, a small excess of air is required to ensure complete combustion. The methods of obtaining these proportions, also the amount of heat liberated by combustion are dealt with later in connection with fuel and combustion. If the heat of combustion (h) is known and the specific heat at constant volume (s_v) for the mixture of air and fuel is also known, then the rise of temperature $= \frac{h}{s_v}$. Unfortunately the value of (s_v) is a variable quantity, increasing with rise of temperature, so that actual rises of temperature are always less than the rise calculated in this way. The value of (s_v) for air is ·169, but the products of combustion particularly at higher temperatures have varying but higher values. What is of more direct practical importance is the rise of pressure. This depends upon compression pressure as well as upon rise of temperature. The previously used formula

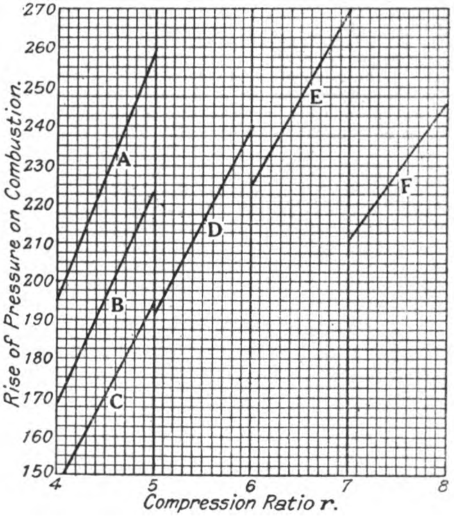
$$\frac{pv}{T} = \text{constant} \quad . \quad . \quad (4)$$

is of use in such calculations.

The rise of pressure in pounds per square inch produced by complete combustion of typical fuels at the various compression pressures obtained by compression ratios between the limits of 4 and 8 is shown on the diagram Fig. 7. For instance, in the case of petrol, when the compression ratio is 4, the rise of pressure is 195, while when the compression ratio is 5, the rise of pressure is 258, and so on at other points. The diagram gives useful average figures. Some fuels require higher compression pressures than others to give the best results while other fuels cannot be used with high compressions. For instance petrol if used with a higher compression than about 5½ causes pre-ignition trouble. The total pressure is the sum of the increase on combustion and the compression pressure which may be obtained from Fig. 5.

Cooling Cylinders. With the exception of some motor cycle and some air-cooled aircraft engines, internal combustion engines are water-cooled. Cooling is a necessary evil; a substantial portion of the heat energy of the fuel is wasted thereby, but it is essential that the cylinders, pistons, etc., should be kept cool enough to work properly. Large engines waste less heat in the cooling water than small, as their capacity is greater in proportion to the area of cylinder wall, etc., to be kept cool.

With small and medium stationary gas engines, cooling tanks are provided. The capacity of the tanks varies from 25–35 gallons per b.h.p. hour. The loss by evaporation is 0·2 to 0·4 gals. per b.h.p. hour. With cooling towers as used with large engines, the evaporation losses are from ·4 to ·5 gals. per b.h.p. hour.



A = Petrol
B = Paraffin
C = Illuminating gas—650B.T.U per cubic foot
D = Natural gas—1,000 B.T.U. per cubic foot
E = Producer gas—150B.T.U. per cubic foot
F = Blast furnace gas—100 B.T.U. per cubic foot
F = Blast furnace gas—100

FIG. 7.

A usual rise of temperature of cooling water in its passage through the cylinder jackets is 40° Fahrenheit, although considerable variation in this figure is permissible. This requires about 7 gallons per b.h.p. per hour through the engine.

The following table gives cooling data for typical engines—

	B.h.p.	Rise of temperature.	Inlet temperature.	Gallons per b.h.p. hour through jackets.	B.th.u. lost per b.h.p. hour
Motor-car petrol engine . .	15–75	40° F.	160° F.	7·5	3,000
Diesel engines .	360	49° F.	54° F.	5·6	2,750
Gas engine, single cylinder . .	140	80° F.	60° F.	3·5	2,800
Gas engine large, vertical . .	750	42° F.	62° F.	6·5	2,730

The percentage of the energy of the fuel which is lost in the cooling water is from 25 to 40 per cent.

DESIGN RULES.

Internal combustion engines are designed and dimensioned in accordance with recognized engineering principles modified as experience in specific types shows desirable or permissible. A few general rules will, however, be found useful here.

Cylinders. Assume maximum pressure in cylinders of 350 lbs. per square inch. Inside diameter $= d$ inches. Thickness of inner wall or liner $= t$ inches.

$$\text{Then } t = \cdot047d + \cdot375 \quad . \quad . \quad . \quad (9)$$

The thickness of the wall may be reduced gradually towards the open end, as the internal pressure is less there, until it becomes

$$t = \cdot02d + \cdot5 \quad . \quad . \quad . \quad . \quad . \quad (10)$$

These rules do not apply to Diesel engines or to those fuel injection engines where the maximum pressure is greater than 350 lbs. per square inch. Nor do they apply to motor car and aircraft engines. In this case when cast iron is used, the thickness varies from 4 to 8 mm. according to dimensions. The thickness at the combustion end is greater than that at the crank case end. Cast iron is in general use for cylinders, but steel cylinders or liners are largely employed in aircraft engines and sometimes in motor-car engines.

Fly-wheels. The turning moment in internal combustion engines is so extremely variable, particularly in single-cylinder engines, that heavy fly-wheels must be provided to keep the "cyclic irregularity," that is the variation from maximum to minimum angular velocity of the crankshaft during the cycle, within limits determined by the class of work done. This cyclic irregularity is denoted by the ratio

$$\frac{\text{variation of velocity}}{\text{mean velocity}} = \frac{1}{m}$$

m varies from 30 in ordinary workshops to 300 in electrical plant.

Cast-iron fly-wheels should have a rim velocity not exceeding 100 ft. per second. If b = breadth of rim and R and r its outer and inner radii, all in feet, and n = revs. per minute,

$$\text{Kinetic energy} = \cdot 1205\,(R - r)\,b.n \text{ foot lbs.} \qquad (11)$$

Valves. "Poppet" or "mushroom" inlet and exhaust valves are almost universally employed. Even when there is no obstruction, such as a valve stem, the lift should not exceed one-fourth the diameter of the opening. If d = diameter of opening and l = lift of valve,

Area of valve − area of stem, etc. = area of opening round periphery of valve ;
or area of valve − area of stem, etc. $= \pi\, dl$.

If area of stem, etc., is neglected, $l = \dfrac{d}{4}$

Sizes of valves depend upon velocity of gas, which again depends upon speed, dimensions and pressures in engine. The valves should be of such dimensions that the mean velocity of the gas through them is between 5,000 and 6,000 ft. per minute, although in motor car engines velocities as high as 8,000 ft. per minute occur. If V = mean velocity through inlet valve, D = diameter of cylinder, n = revs. per minute, and s = stroke in inches,

$$V = \frac{D^2 \cdot n \cdot s}{6d^2} \qquad (12)$$

FUEL AND COMBUSTION.

Fuels for internal combustion engines consist of various chemical compounds of carbon (C) and hydrogen (H), but for combustion calculations they may all be regarded as containing merely certain percentages of C and H, and in some cases oxygen (O).

Data required for calculation—

	Hydrogen.	*Carbon.*	*Oxygen.*	*Nitrogen.*
Atomic weights .	1	12	16	14
Molecular weights .	2	12	32	28
Formula for atom .	H	C	O	N
Formula for molecule	H_2	C	O_2	N_2

Formulae for oxygen and air required for combustion—

$$\begin{aligned} 2\,H_2 + O_2 &= 2\,H_2O \text{ (steam or water)} && (13) \\ 2 \times 2 + 32 &= 2\,(2 + 16) \\ \text{or 1 lb. H} + 8 \text{ lbs. O} &= 9 \text{ lbs. } H_2O \end{aligned}$$

$$\begin{aligned} C + O_2 &= CO_2 && (14) \\ 12 + 32 &= 12 + 32 \\ \text{or 1 lb. C} + 2\tfrac{2}{3} \text{ lbs. O} &= 3\tfrac{2}{3} \text{ lbs. } CO_2 \end{aligned}$$

$$\begin{aligned} 2\,C + O_2 &= 2\,CO && (15) \\ \text{or 1 lb. C} + 1\tfrac{1}{3} \text{ lbs. O} &= 2\tfrac{1}{3} \text{ lbs. CO} \end{aligned}$$

$$\begin{aligned} 2\,CO + O_2 &= 2\,CO_2 && (16) \\ \text{or 1 lb. CO} + \tfrac{4}{7} \text{ lb. O} &= 1\tfrac{4}{7} \text{ lb. } CO_2 \end{aligned}$$

When oxygen is present in the fuel, a corresponding quantity of hydrogen must be regarded as neutralized and not available for combustion, since it is already combined with some of the hydrogen, forming moisture. If C_w, H_w, and O_w are the weights of the respective elements in a fuel, then the oxygen required for combustion is

$$O = C_w \times 2\tfrac{2}{3} + \left(H_w - \frac{O_w}{8}\right)8 \quad . \quad . \quad . \quad (17)$$

Air contains $\begin{cases} \text{O } (20{\cdot}9\% \text{ by volume or } 23{\cdot}1\% \text{ by weight});\\ \text{N } (79{\cdot}1\% \text{ by volume or } 76{\cdot}9\% \text{ by weight}).\end{cases}$

From which the air required to give the above quantities of O necessary for combustion may be calculated. 4·33 lbs. of air contains 1 lb. of oxygen.

The weight of 1 cubic foot of air at normal temperature and pressure (n.t.p.) is ·0807 lbs. or 12·39 cubic feet weighs 1 lb. Normal temperature is 0° Cent. or 32° Fahr. Normal pressure is 760 mm. or 29·32 ins. of mercury.

As mentioned previously, the pressure (p), volume (v), and absolute temperature (T) of any given quantity of any gas are given by—

$$\frac{pv}{T} = \text{a constant} \quad . \quad . \quad . \quad . \quad (4)$$

To determine the volume (V) at n.t.p. modify formula thus—

$$V = \frac{p.v.\ 273}{760\ (t + 273)} \text{ or } \frac{P.v.\ 460}{29{\cdot}92\ (T + 460)} \quad . \quad (18)$$

where p = pressures in millimetres of mercury,
P = pressures in inches of mercury,
T = temperatures in degrees Fahr.,
t = temperature in degrees Cent.

Heat of Combustion—

1 lb. H burnt to H_2O gives out 60,890 B.Th.U. (higher value)
52,200 B.Th.U. (lower value).
1 lb. C burnt to CO_2 gives out 14,544 B.Th.U.
1 lb. CO burnt to CO_2 gives out 10,200 B.Th.U.

The difference between the higher and lower values in the combustion of hydrogen is equal to the latent heat of the steam formed. The calorific value of coal is given by the equations—

$$X = 60{,}890 \left(H_w - \frac{O_w}{8}\right) + 14{,}544 \times C_w \text{ (higher value) } (19)$$

$$X = 52{,}200 \left(H_w - \frac{O_w}{8}\right) + 14{,}544 \times C_w \text{ (lower value) } (20)$$

It has been found necessary to modify the constant for carbon in oil fuels since carbon may then have different calorific values depending upon the nature of its compounds. Equations (19) and (20) will then be modified for oil fuel as follows—

$$X1 = 60{,}890\left(H_w - \frac{O_w}{8}\right) + 13{,}500 \times C_w \text{ (higher value)} \quad (21)$$

$$X1 = 52{,}200\left(H_w - \frac{O_w}{8}\right) + 13{,}500 \times C_w \text{ (lower value)} \quad (22)$$

For gaseous fuels the quantities of the various constituents must be known and the calorific value of each worked out independently. The calorific values of some constituents such as methane and ethylene are different from the calorific values obtained by calculation from the elements, since some are exothermic, that is, evolve heat on formation, and others are endothermic, that is, absorb heat on formation. The converse action takes place when the compounds are split up for combustion. Exothermic and endothermic effects are negligible in all but gaseous fuels.

Engines adapted to use petrol or paraffin or heavy oils develop power in respective approximate proportions 10 : 9 : 8.

Fuel.	Ratios of Molecular Weights or Sp. Gr. (H = 1).	Wt. : O for combustion of 1 unit.	Wt. : Air for combustion of 1 unit.	Calorific Value.	
				Higher.	Lower.
H. Hydrogen (burnt to H_2O)	1	8	34·8	60,890	52,200
C. Carbon (burnt to Co_2 . .	12	2·67	11·6	14,544	14,544
Carbon (burnt to CO) . .		1·33	5·8	4,344	4,344
O. Oxygen . .	16	—	—		
N. Nitrogen. .	14	—	—		
H_2O Water or steam	9	—	—	—	—
CO_2. Carbon dioxide	22	—	—	—	—
CO. Carbon monoxide (burnt to CO_2) .	14	57	2·48	10,200	10,200
CH_4. Methane or marsh gas (burnt to H_2O and CO_2) .	8	4	17·4	24,000	21,820
C_2H_4. Ethylene (burnt to H_2O and CO_2) . .	14	3·43	14·9	21,900	20,660

A specimen petroleum contains 85½ per cent carbon; 13½ per cent hydrogen; 1 per cent oxygen. Calorific value by the formula (19) is 20,430 (higher value), or by formula (20) is 19,300 (lower value).

This neglects exothermic and endothermic values. The actual mean calorific value of such a fuel with the three main elements in the proportions given above is 19,600 B.Th.U. per pound.

The power developed by an engine depends upon its height above sea level since the rarity of the air at the higher altitudes results in a smaller mass or weight of air or mixture being drawn into the cylinder so that less power is developed. The decrease of power for each 1,000 feet above sea level may be taken as 4 per cent of the power at sea level.

TYPES OF ENGINES.

As mentioned above, all gas engines fall broadly under one of the two types : (*a*) constant volume ; (*b*) constant pressure, quite independently of whether they follow the four or two-stroke cycle. The following are various types of engines—

1. Gas Engines.

These are of the constant volume type ; that is, a mixture of air with coal gas, producer gas, blast furnace gas, or natural gas, is compressed and ignited at constant volume. In this section are found the following : the ordinary single-cylinder horizontal engine in all sizes from ½ h.p. to about 170 h.p. ; horizontal tandem engines, vertical tandem engines with as many as six pairs of cylinders, double-acting engines both horizontal such as large M.A.N., or vertical two-stroke engines with two oppositely moving pistons in the same cylinder such as the Oechelhauser, the Fullagar and the Junkers ; also the four-stroke and two-stroke Humphrey explosion pumps in which the piston is replaced by the surface of a moving column of water.

2. Petrol Engines.

In such engines a mixture of air and vapour from an easily vaporized fuel, such as petrol or benzol, is compressed and ignited, with the result that there is practically instantaneous combustion, so that this type of engine is also of the constant-volume engine. Most gas engines can be run on petrol, though it is much more expensive than gas. Motor-car engines, air-craft engines, some small marine engines, engines for small electric lighting and similar plant, are included under this heading.

3. Diesel Engines.

In these engines, which are of the constant-pressure type, pure air is compressed but to a much higher pressure than in the preceding types. The fuel is then injected by means of compressed air during the early part of the working stroke, and burns while the piston is moving outwards, so that the pressure does not then substantially differ from the compression pressure. After the injection period, the working fluid expands as usual.

4. OIL-INJECTION ENGINES.

These engines, sometimes referred to as semi-Diesel, differ from Diesel engines, broadly speaking, in two ways only—the oil is injected by a pump and not by compressed air, and secondly the compression is not as high, so that a "hot bulb" has to be utilized for ignition. These distinctions do not always apply, the boundary between the two types being somewhat indefinite.

5. PARAFFIN ENGINES.

These engines fall under the constant-volume class, since a mixture of air and paraffin vapour is compressed and ignited by a spark, or by putting the compressed mixture into communication with a hot chamber. These engines are made of a horizontal single-cylinder stationary type or of the multi-cylinder motor-car type, the former being generally ignited by opening a valve which communicates with a highly heated chamber, and the latter by electric ignition. The paraffin engine differs from the petrol engine in that a specially heated vaporizer for the paraffin is required.

SINGLE-CYLINDER GAS ENGINES.

The ordinary single-cylinder gas engine is probably in more general use for stationary power purposes than any other type of internal combustion engine. It requires no special vaporizing or injection devices, and is accordingly one of the simplest engines. Town gas is generally used for small engines, but its cost is rather high, particularly in districts where the gas works is small and the area of supply large. Producer gas is cheaper, particularly in the larger sizes, but is not so satisfactory in working, as even with the most careful cleaning and drying, the engines using it suffer from clogging of such parts as ignition points, valves, and pistons much more than when using town gas. The calorific value is also less and higher compressions are desirable.

Crossley Engine. Fig. 8 shows a longitudinal section through an engine which is made in sizes varying from 70 to 150 b.h.p., while Fig. 9 shows a transverse section through the inlet valve and governing gear. The crank shaft is mounted in three bearings—one on each side of the crank and a third on the far side of the fly-wheel. The trunk piston 24 is long, giving a good bearing surface against the cylinder walls, and is fitted with a number of rings—in this case, seven. The bedplate A is bolted to a concrete foundation, and is cast with an upper extension B, which encloses the cylinder liner C, the space between the two forming a water-jacket. A groove is formed round the base of the bedplate to catch any waste oil and prevent damage to the concrete foundation. The breech end or combustion chamber U is bolted to the end of the bedplate casting, and encloses in its upper part the mixture and gas admission valves 1, 3, and in its lower part the exhaust valve F. The combustion chamber and the exhaust pipe G

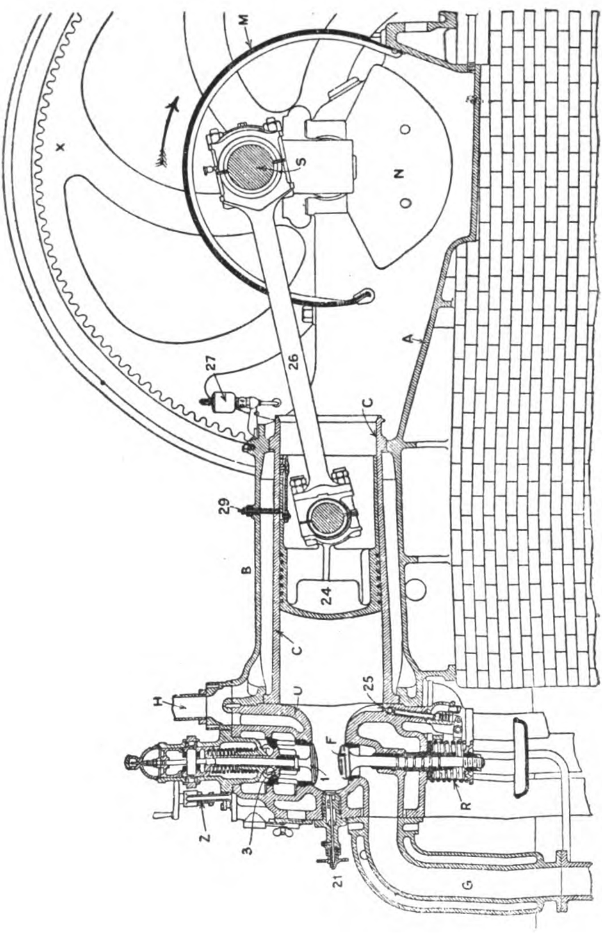


FIG. 8.

are cooled by means of a water-jacket. Water enters at the lowest part of the jackets, and leaves through the upper pipe H. The design avoids overhang of the cylinder.

The side shaft or cam shaft is driven at half crank-shaft speed through skew gearing, and it is provided with two case-hardened steel cams. The exhaust cam acts on the stem of the exhaust valve F through a rocking lever, the other end of which lifts the valve against the resistance of a spring R. The exhaust roller 8 can be shifted slightly on its pin, so that it will engage also, when required, a narrow starting or half-compression cam. The exhaust valve F is then, in addition to its ordinary lift, raised during the early part of the compression stroke, so as to reduce the compression resistance at starting.

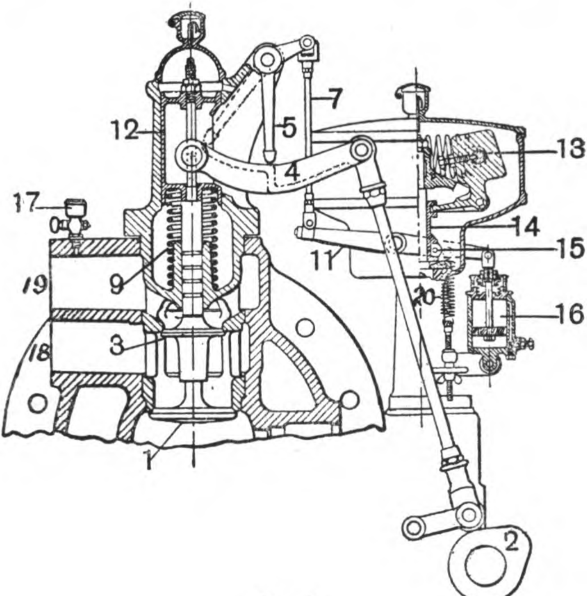


FIG. 9.

The engine is controlled by a centrifugal governor 13, driven from the cam-shaft, in conjunction with a gas valve 3 and a mixture valve 1. The inlet valves are operated from the inlet cam 2 through a push rod and rocking lever 4, which is adapted to turn about the lower end of the fulcrum lever 5. The mixture valve 1 and the gas valve 3 are in this way opened to an extent depending upon the position of the fulcrum lever 5, which is adjusted by means of a governor designed to allow a variation of speed of from 1½ to 2 per cent above or below normal from no load to full load. The lever 4 only engages the fulcrum lever 5 when it is lifted by the cam; at other times there is a very small clearance between the two, which allows the governor to perform any necessary adjustment of the fulcrum lever 5; but when the lever 4 is being rocked by the cam, the pressure between it and the fulcrum lever 5 holds the latter stationary. The normal speed may be varied slightly by adjustment of the spring 20.

To prevent hunting of the governor, a dashpot 16 is arranged at the end of the lever 11.

In some smaller Crossley engines the inlet valve 2 (*see* Fig. 10) has a constant lift and the governor acts on a throttle valve 21. These engines are made in sizes developing from 7 to 12½ h.p. and may be fitted with either hot tube or electric ignition. The general arrangement is similar to that of the larger engines previously described. The crank-shaft runs in two bearings only, and two outside fly-wheels are provided.

Premier Engine. In this engine the enclosed centrifugal governor is disposed vertically at the crank-shaft end of the engine, and is driven directly from the crank-shaft through skew gearing, the object being to avoid driving through the cam-shaft so that the effect of the irregular cam operation is not felt by the governor. The power is regulated by varying the lift of the gas and air inlet valves, the valves, valve gearing, and combustion chamber being shown in section in Fig. 11. The gas valve C is formed with a ring, which controls the opening of the air ports P, and these parts are mounted on the spindle of the mixture valve D, so that they can move slightly on the spindle under the action of a spring. This ensures that both valves engage their seatings properly. It will thus be seen that the variation of the lift or stroke of the valves regulates (*a*) the quantity of gas passing the gas valve C, (*b*) the quantity of air admitted through the ports P, and (*c*) the quantity of mixture passing the valve D.

The valve spindle is depressed by means of the cam E, through the medium of the notched plate H pivoted at I, the rod F, and the rocking lever G.

The stroke of the admission valves is controlled by the governor, which turns the spindle J and swings the rod F in such a manner that, when the speed of the engine increases, the hardened end of the rod F engages the hardened notched plate H nearer to its fulcrum I, and thus a smaller charge is admitted When the rod F is moved sufficiently far to the left it misses the notched plate H entirely, and no gas or air is admitted to the cylinder. This arrangement constitutes a combination of (*a*) variable mixture or throttling control, and (*b*) hit-and-miss control. The latter method operates only when the engine is very lightly loaded or possibly running without load; the variable mixture method of control would then be uneconomical and would possibly lead to missed explosions. The exhaust valve is operated from the cam shaft through the rocking lever

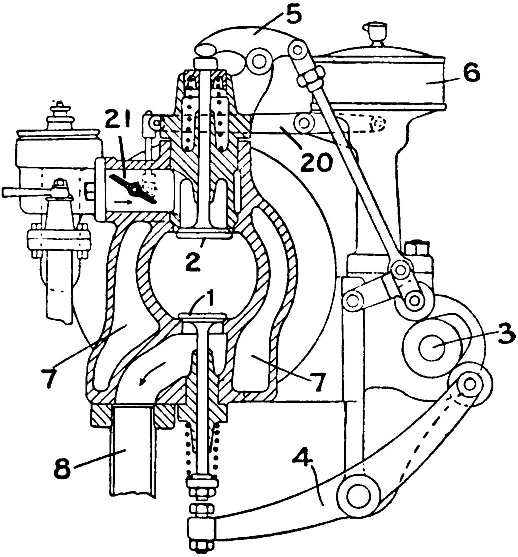
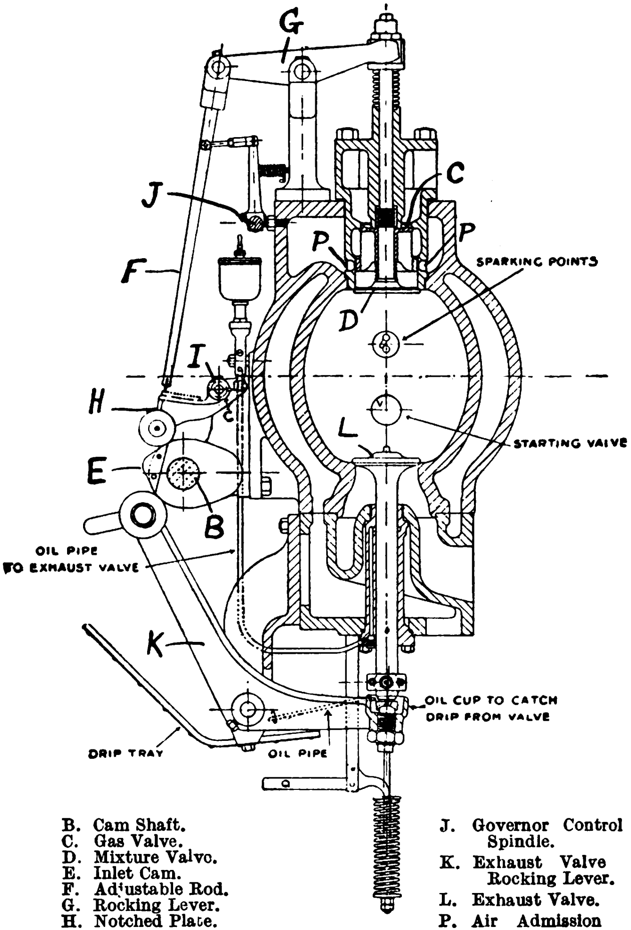


FIG. 10.—CROSSLEY SMALL GAS ENGINE. VALVES AND GOVERNING GEAR.

1. Exhaust Valve.
2. Inlet Valve.
3. Cam Shaft.
4. Exhaust Valve Rocking Lever.
5. Inlet Valve Rocking Lever
6. Governor.
7. Water Jackets.
8. Exhaust Pipe.
20. Governor Lever.
21. Throttle Valve.

K, and the valve itself in the larger sizes is prevented from burning by water circulated inside it, which also renders it possible to lubricate the stem of the valve as shown. Each of the valves is provided with two springs, and, in the event of breakage of either spring, the other will keep the engine going while a new spring is put in.

Robson Engine. In the Robson engine the power of the engine is controlled by the governor by mechanism which can act on the variable "quality" principle or the "hit-and-miss"



B. Cam Shaft.
C. Gas Valve.
D. Mixture Valve.
E. Inlet Cam.
F. Adjustable Rod.
G. Rocking Lever.
H. Notched Plate.
I. Pivot of Notched Plate.
J. Governor Control Spindle.
K. Exhaust Valve Rocking Lever.
L. Exhaust Valve.
P. Air Admission Ports.

FIG. 11.—PREMIER GAS ENGINE. VALVE GEAR.

principle. Fig. 12 shows the inlet mechanism as adapted for quality governing. The gas valve *a* is given a variable lift controlled by the governor. Air is admitted by the lower inlet passage and the mixture is admitted to the cylinder by the valve *b*, which is given a constant lift from a rocking lever 1. The long stem *e* of the valve *b* is pulled upwards by a spring *h*.

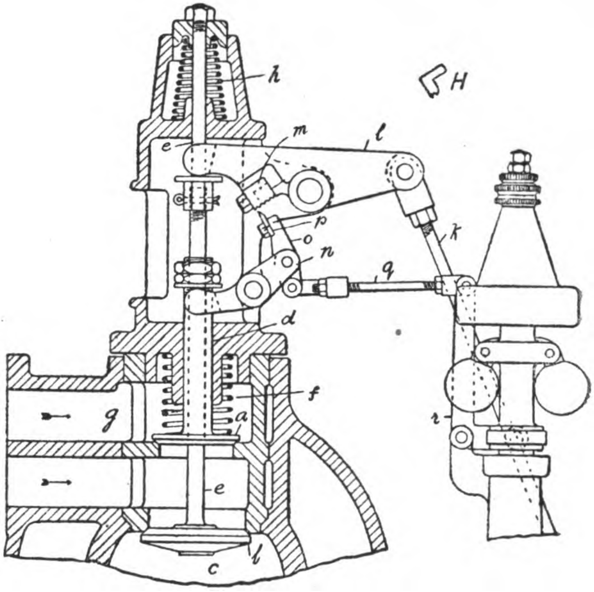


FIG. 12.—ROBSON ADMISSION VALVES AND CONTROLLING MECHANISM.

a. Gas Valve.
b. Mixture Valve.
c. Engine Cylinder.
d. Stem of Gas Valve *a*.
e. Stem of Valve *b*.
f. Spring of Valve *a*.
g. Gas Passage.
h. Spring of Valve *b*.
l. Rocking Lever for Valve *b*.
m. Stepped Block.
n. Rocking Lever for Valve *a*.
p. Pecker.
r, *q*, *o*. Governor Connections.

The gas valve *a* has a hollow stem *d*, which slides on the stem *e* of the mixture valve *b*; it is forced downwards by a spring *f*.

The lift of the gas valve *a* is controlled by the governor by means of the mechanism now to be described. The upper end of the hollow valve stem is raised by the left arm of the rocking lever *n*, the right arm of which carries hinged to it a short finger *o* connected to the governor at its lower end by the link *q*. The upper end of the finger *o* carries a "pecker" *p*, which consists of a hardened steel piece with a V edge. This pecker is engaged by any one of several steps or notches in a stepped block *m* carried by the rocking lever 1. The movement of the rocking lever 1 thus depresses the pecker, and so operates the rocking lever *n* and lifts the gas valve *a*. The vertical arm *r* of a bell crank lever forming part of the governor gear adjusts the link *q* according to the speed, and so tilts the rocking lever *o* that the pecker *p* engages the block *m* closer to or further from the fulcrum of the lever 1. When the speed of the governor increases beyond the desired amount the bell-crank lever *r* tilts the finger *o* so as to move the pecker further to the right, with the result that the finger is depressed at a later moment in the cycle of operations, owing to the inclination of the stepped face of the block *m*. The opening of the gas valve is consequently delayed and the lift reduced, and the charge admitted to the cylinder is weakened. There is no throttling action ; a full charge of air enters the cylinder under all conditions.

The mechanism can be adapted for "hit-and-miss" governing by replacing the stepped block *m* by a block having one groove only, of the shape shown at H. With the ordinary load the pecker is engaged by the groove every time it descends, but with increase of speed the pecker is moved by the governor slightly to the right, so that it misses the groove, the gas valve is not lifted, and a full charge of pure air is admitted to the cylinder.

Starting. As every working stroke in an internal combustion engine depends upon the previous stroke, starting is an inherent difficulty, independent means for turning being always required. In the smaller sizes the engine can be turned directly by means of the fly-wheel, but in the larger sizes it is necessary (*a*) to pump an explosive charge into the cylinder, or (*b*) to work the engine by admitting compressed air during the working stroke until the necessary charge is drawn in and fired in the ordinary way, or (*c*) to turn the engine by a small independent motor, such as a petrol engine or an electric motor. In both methods (*a*) and (*b*) the crank shaft must first be barred or otherwise turned into such a position that the piston has covered about one-third of the working stroke.

LARGE GAS ENGINES.

The single-cylinder gas engines just described are seldom made in larger sizes than about 150 h.p. Greater powers are obtained as follows—

(*a*) Two, three, or four horizontal cylinders are coupled together so as to work on the same crank shaft.

(*b*) The engines are made double-acting.

(*c*) The two-stroke cycle is adopted.

(*d*) Two horizontal cylinders are placed tandem-fashion—that is, one cylinder is arranged behind and co-axial with the other, the two pistons having a common piston-rod connected to a single crank.

(*e*) Two pistons are placed in the same cylinder.

(*f*) Two, three, four, or six vertical cylinders are arranged to work on the same crank shaft.

(*g*) Several pairs of tandem vertical cylinders are arranged to work on the same crankshaft.

The principal difficulty in the use of large cylinders is the cooling of the cylinder and other parts, particularly the piston and exhaust valve, since the cooling area increases with the square of the linear dimensions while the volume of the hot products increases as the cube. The mere jacketing of the cylinder and combustion chamber is insufficient and arrangements have generally to be made to circulate cooling water through the piston and exhaust valve.

A selection of examples will now be described.

Premier Multi-cylinder Scavenging Engines. This engine falls under heading (*a*), and in addition possesses unusual features which tend to a high thermal efficiency and larger cylinder sizes. The products of combustion are swept out by a charge of air forced through the clearance space so that the incoming mixture of gas and air is not contaminated by the exhaust gases. The cylinder is thus scavenged and also better cooled. These engines are made in sizes developing from 170 to 270 h.p. per cylinder and are constructed with two, three, or four cylinders side by side, each working on its own crank on a common crank shaft. Fig. 13 shows a longitudinal section through one cylinder of a four-cylinder engine. With the exception of a scavenging charge the cycle is the usual four-stroke one. At the beginning of the suction-stroke the admission valve E will be open, exhaust valve H nearly shut, and the annular gas valve G shut, but opening when the piston has moved a short distance on the suction stroke. The air supply is admitted through light lift valves F. The piston is of the differential type and therefore when it moves forward the air entering through the valves F will divide up, a portion entering the enlarged cylinder N by way of the port D, and another portion going through the passage C, ports P, and valve E to the motor cylinder Z. The gas is fed through a gas cock to the space B, whence it passes through the annular gas valve G, past the air ports PP, where it meets and mixes with the entering air. Towards the end of the stroke the gas valve is closed, and shortly thereafter the admission valve E also.

On the compression stroke the air trapped between the two pistons is compressed idly to a pressure of about 5 lbs. per square inch, the compression pressure in the inner cylinder being from 100 to 180 lbs., depending upon the clearance volume, which again depends upon the kind of gas which the engine is designed to use. On the working stroke the scavenging air expands

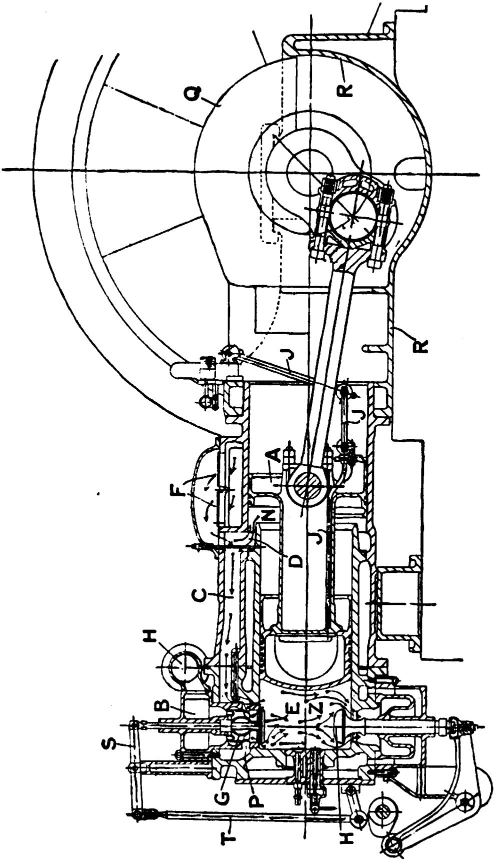


Fig. 13.

again. On the exhaust stroke the scavenging air is recompressed till the crank reaches the position shown in the diagram when the main admission valve E opens and the scavenging air enters and clears out the products of combustion. A considerable excess of air is supplied which passes out through the exhaust valve.

Each piston is coupled to a separate crank. A massive bed R carries all the crank shaft bearings and to it also the cylinder jackets are bolted. The cylinders are not, however, overhung but are supported by a transverse member of inverted channel section, the end of which is shown in Fig. 13. Rigid support of the ends of the cylinders is essential, since they carry the bearings for the cam shaft K, the gearing and casing for driving the cam shaft, and the bearings for the exhaust valve rocking levers. The engine rests upon a heavy concrete foundation. The cranks are completely enclosed by sheet metal oil guards Q. The common cam shaft K runs right across the end of the engine and is driven by a shaft V arranged alongside the end cylinder. The exhaust valves, which are as usual underneath, are operated by rocking levers and the overhead inlet valves through push rods T and rocking levers S. The stroke of the inlet valves is constant and regulation of the engine by the governor is effected by means of a throttle valve in the gas supply to the pipe H, which runs across the four cylinders and communicates with each. The governor can be connected to the air throttle valve also, but the makers find that with ordinary producer gas it is preferable to vary the gas only. An air throttle controls the air supply by hand to all four cylinders, and in addition each cylinder may be provided with a separate throttle valve. The governor U is arranged on a vertical shaft, and is driven by skew gearing from the crank shaft, the arrangement being similar to that described previously in connection with the Premier single-cylinder gas engine. Each of the inlet valves in this engine is provided with two springs W, either of which can be changed while the engine is running, so that a broken spring does not cause a stoppage. An independent magneto M is fitted to each cylinder, each magneto being operated by its own rod from an eccentric on the cam shaft.

The engines are started by compressed air which is supplied automatically to the middle two of the four cylinders. The gear consists of a distributing valve which is operated by a cam on the main cam shaft K, and admits compressed air to the two cylinders at the proper intervals. It can be put into and out of action readily by a hand lever whatever the position of the engine, and should it be left in gear after the engine is running normally, no harm will be done or air wasted, since admission of air stops as soon as the cylinder explosions commence, owing to the explosion pressure being greater than the pressure of the starting air.

The differential piston is cooled by water supplied through jointed pipes J, there being an inflow and an outflow pipe to each piston. The main bearings and crank shaft may be flooded with oil by a pump or may be supplied with oil under pressure. The cylinders, gudgeon pins, and exhaust valves are supplied with measured quantities of oil forced to the points at which it is required by means of plunger pumps operated in a positive manner from the cam shaft through a rod Z by means of pawl and ratchet gear.

The Premier Company also manufacture engines working on the same general principle but having two single-acting tandem cylinders, the working strokes being alternate so as to obtain one impulse per revolution on each crank. Both cylinders of each pair are scavenged from a single-acting pump worked by a separate connecting rod from an upward projection on the main big end. The pump supplies its charge of air to the cylinders alternately.

Oechelhauser Two-stroke Engine. In this engine power is obtained from a single cylinder by the adoption of the two-stroke cycle, and by the use of two pistons in the one cylinder.

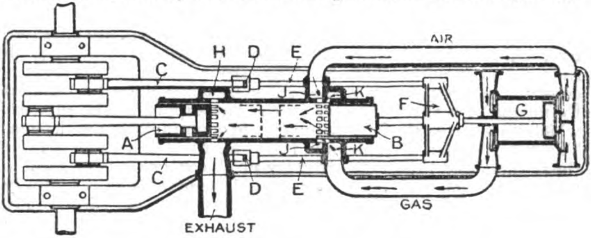


FIG. 14.

Messrs. Beardmore, of Glasgow, have manufactured engines of various sizes developing from 400 to 1,500 h.p. in one cylinder. The 1,500 h.p. engine has a cylinder 42 ins. in diameter, and each piston has a stroke of 51 ins. A fly-wheel suitable for use when the engine drives a dynamo would have a diameter of 20 feet and a weight of 100 tons, and the overall length of the engine, including the fly-wheel, would be 56 feet. The weight of the engine without the fly-wheel is 180 tons. Where two engines of this size are coupled together so as to work on one crank shaft the cylinders act alternately and a single fly-wheel weighing only 70 tons is placed on the crank shaft between them.

A diagrammatic sectional plan of an engine is shown in Fig. 14. Both pistons move towards and away from one another simultaneously, the front piston A being connected directly to the middle crank, while the back piston B is driven from the two outer cranks through two connecting rods C, cross-heads D, side rods E, and a back cross-head F. The piston-rod for the back piston B is also extended to work a double-acting pump G, the inner end of which supplies gas to the main cylinder, while the outer end supplies air.

There are no valves in this engine; the burnt gases are exhausted, and the air and gas admitted through ports which are uncovered by the pistons themselves when the latter are in their outer-most position. The inner piston A controls a ring of exhaust ports. The outer piston B controls (1) a ring of air ports J, and (2) another ring of gas ports K slightly further out. The scavenging air when first admitted is at a pressure of from 5 to 7 lbs. per square inch above atmospheric pressure. When exhausting and charging, which operations take place simultaneously, it is intended that the burnt gases and entering mixture should be separated by a layer of scavenging air, this action being sometimes described as stratification.

M. A. N. Tandem Double-acting Engine. Although the theoretical value of large gas engines from the point of view of economy are well known in this country, especially when using blast furnace or coke oven gas, little progress has been made in their use as compared with that made on the Continent, particularly in Belgium and Germany.

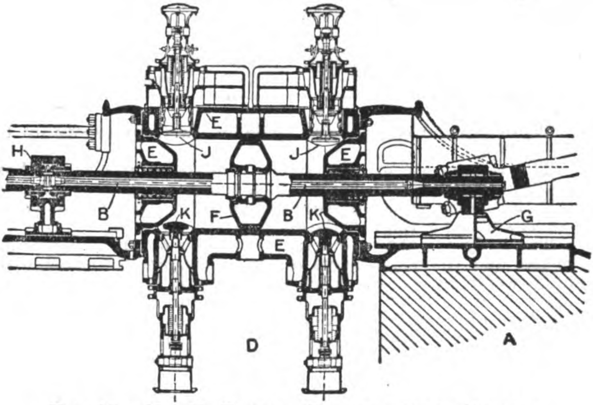


FIG. 15.—M.A.N. TANDEM DOUBLE-ACTING ENGINE.

A. Concrete Foundation.
B. Piston-rod.
D. Pit for Exhaust Valve Gear.
E. Water Jacket.
F. Piston.
G. Crossheads.
H. Slide for Supporting Pistons
J. Inlet Valves.
K. Exhaust Valves.

Fig. 15 shows a sectional elevation through one cylinder of a tandem double-acting four-stroke engine, the crank shaft (not shown) being arranged on the right and a similar cylinder (also not shown) on the left, its piston being connected to the hollow piston-rod. The two cylinders work together in such a way every stroke has one power impulse, there being th ing strokes in the complete cycle of two revolutio ke for each end of each cylinder. The sequence of ope lustrated in the following table—

Revolution.	Stroke.	BACK CYLINDER.		FRONT CYL	
		Front End.	Back End.	Front End.	Ba
First.	Forward.	Suction.	Compression.	Working.	Ex
	Return.	Compression.	Working.	Exhaust.	Su
Second.	Forward.	Working.	Exhaust.	Suction.	Co si
	Return.	Exhaust.	Suction.	Compression.	Wo

These tandem engines are made in sizes developing 3,000 h.p.

The framework of the engine consists of a massiv casting which is bedded down to a concrete foundation cylinders being arranged over a pit D, so that the exhaust stems are below the floor level.

The body of each cylinder, the cylinder covers and s boxes, and the valve cages and seatings, are furnishe large water jackets E. The piston rods B and the pist are hollow and have water circulating through them co ously at a pressure of from 45 to 65 lbs. per square inch. provision for cooling is very important, in view of the large size of these cylinders, viz., 4 to 5 feet in diameter. The weight of the pistons and rods is entirely taken b cross-head G, the slide H between the two cylinders, a further slide (not shown) on the extreme left of the engi that the pistons float in their cylinders.

The inlet valves J (on top) and the exhaust valves K (b are operated by a very simple gear, in which one ecc only is employed and no cams. This gear is shown in Fi The eccentric L on the half-speed shaft reciprocates eccentric-rod M, which is connected at its lower end to a ro lever N. This lever lifts the exhaust valve K through medium of a pivoted lever O, which engages the lower e the valve stem. Although the rocking lever N recipro continually, it only lifts the valve when it is at the upper of its movement. During the other part of its movemen circular end of the lever, which is concentric with the pive merely rubs against the lever O without lifting it.

The inlet valve J and the sliding sleeve or hollow cylind valves are rigidly connected and are operated by ro rs worked by a rod T from the eccentric L. The r

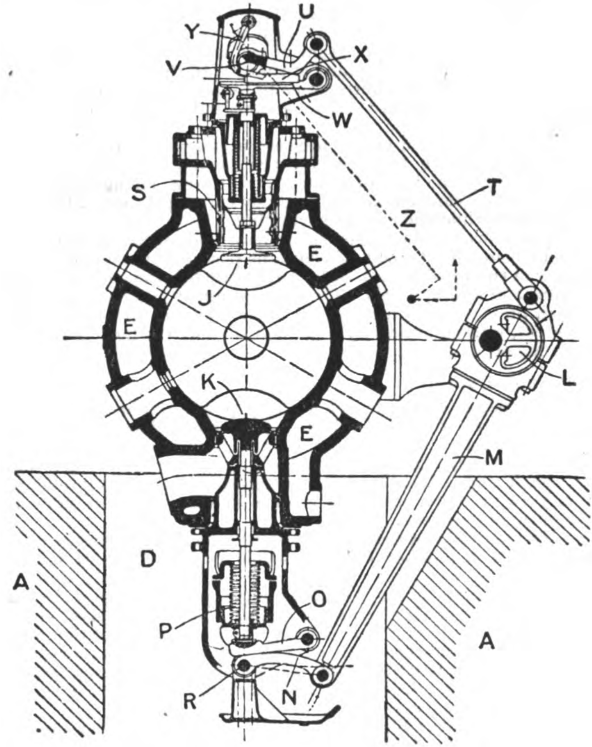


FIG. 16.—M.A.N. VALVE GEAR.

A. Concrete Foundation.
D. Pit for Exhaust Valve Gear.
E. Water Jackets.
J. Inlet Valves.
K. Exhaust Valves.
L. Eccentric.
M. Eccentric-rod.

N. Rocking Lever.
P. Exhaust Valve Spring.
S. Air Valve.
T. Inlet Operating-rod.
U, W. Inlet Operating Lever
X. Adjustable Piece.
Y Z. Connections to Governor.

the bent lever U, which is pivoted at V and depresses
ng lever W when required through the medium of an
le piece X. The lever W acts on the end of the valve
d depresses the combined valves, so as to open them
quired. The position of the adjustable piece X deter-
e amount of opening of the valves, the necessary adjust-
eing effected automatically by a bent lever Y connected
governor, and the connections indicated by the dotted

composition of the mixture is practically constant once
lves have been set for the most efficient combustion
ng to the kind of gas used. If the gas changes, the ratio
gas can be readily altered by turning the air slide valves
while the engine is running.
starting the engine compressed air is used, the crank
being first turned to a suitable position by manual barring
the smaller sizes, or by an electric barring gear in engines
500 h.p. upwards. Compressed air at 300 lbs. per square
essure is then admitted to the cylinder through starting
worked at first by hand and then by the half-speed shaft
eans of a cam.

ENGINES.

petrol engine is not a cheap power producer, but in its
sphere it is unrivalled. With its carburetter, ignition
and petrol tank it forms a complete power unit which can
at very high speeds, so that its power-weight ratio is high.
engines can be run at a speed of 3,000 revolutions per
or more, although such a speed is not advisable for
periods.
majority of petrol engines work on the four-stroke cycle,
a number of motor-cycle and motor-boat engines
two-stroke cycle. The valve-operating gear in
use employs cams as in gas and oil engines, the only
used alternative being the sleeve-valve engine
the first instance by the Daimler Motor Company.
gines are both air and water-cooled, the air-cooled
used mostly on motor-cycles and in rotary cylinder
engines, such as the Gnome; water-cooled engines
universally used on motor cars.
the high speeds usual very careful provision has
for lubrication. Broadly, there are two systems
multi-cylinder engines, viz., forced and trough. In
drawn from the sump or lower part of the crank
separate tank is forced under pressure by a pump
engine through passages or pipes to the main
earings and thence through a drilled crank-shaft
ecting rod end bearings. The small end bearings,
walls, and the cams, cam-shaft bearings, and
erally lubricated by the surplus oil splashed out
The Star engine, to be described shortly,
this system.

In the trough or splash system dippers on the lower ends of the connecting rods dip into troughs containing oil kept at a constant level. The troughs are supplied by a pump, and the excess overflows into the sump. The amount of oil thrown about can be regulated to some extent by varying the oil level in the troughs or the lengths of the dippers. The Daimler engine, to be described later, employs this system.

Star Four-cylinder Engines. An example of a modern high-speed long-stroke engine is shown in Figs. 17, 18, the cylinders having a bore of 80 mm. and a stroke of 150 mm. The crank shaft is supported in two end bearings A and a middle bearing B, this being the bearing arrangement in most common use, although five-bearing and two-bearing crank shafts are largely used. The cranks all lie in the same plane, the two end cranks C being coincident.

The question of balance is very important in these high-speed engines with light supports or foundations. With this crank arrangement the primary vertical reciprocating forces are balanced, there being however a secondary vertical vibration due to the length of the connecting rods. This secondary vibration has twice the periodicity of the crank shaft. It has been proposed by Mr. Lanchester that it should be balanced by two masses rotating oppositely at twice crank shaft speed so as to balance one another transversely and balance the secondary vibration of the engine vertically.

The pistons are made exceptionally light, the weight of each piston complete with hollow gudgeon pin F and rings being about 1 lb. 9 ozs. The cylinders are cast in pairs with their water jackets H and valve spaces. The two cylinder castings are bolted to a two-part crank casing cast in aluminium, the two parts of which are divided on a plane containing the axis of the crank shaft. The crank shaft bearings A, B, are bolted up to the upper part. A rotary pump K circulates the water continuously in this engine, although thermo-syphon or natural circulation through a radiator raised slightly above the level of the engine is also largely relied upon.

The cam shaft L is driven from the crank shaft by a silent chain to avoid gear noises and also to obtain a cushioning effect of the cam impulses.

The working parts of the engine are completely enclosed in the crank casing and the valve springs S and the adjustable tappers T are enclosed by detachable cover plates. The crank or big ends of the connecting rods are provided with divided white metal bearings.

It will be seen from Fig. 18 that the axes of the cylinders are "des axe," or slightly out of line with the axis of the crank shaft so as to reduce the side thrust on the cylinder wall during the working stroke.

The lubrication system is of the forced type referred to previously. A pump drains oil from the sump V, and delivers it under pressure through passages W to the three main bearings,

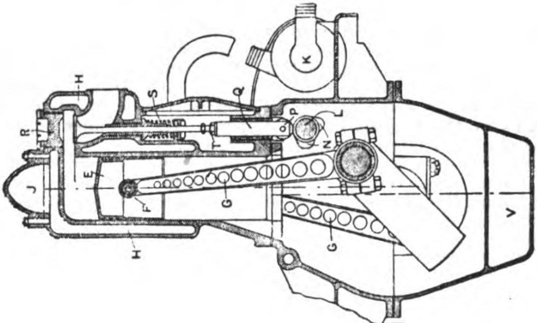


FIG. 18.

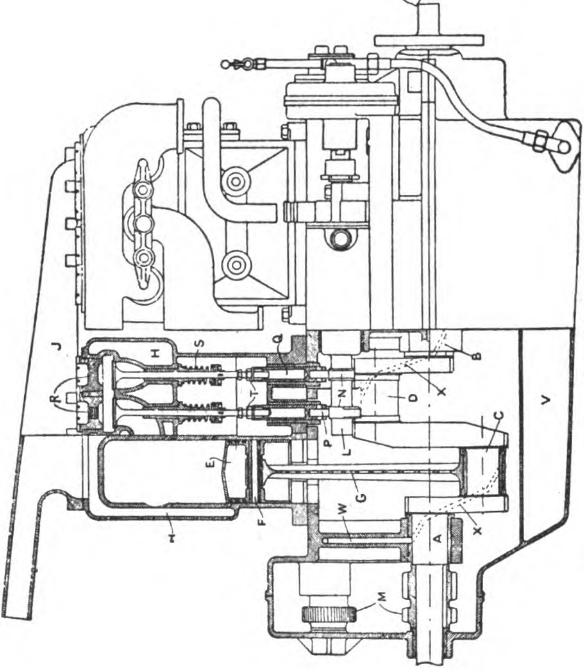


FIG. 17.

from which it passes through the drilled crank shaft to the big end bearings.

Since there is an explosion on alternate down strokes of each cylinder the turning impulses on the shaft occur at intervals of 180°. The cylinders fire in the order 1, 2, 4, 3, or alternatively in the order 1, 3, 4, 2. In the case of a six-cylinder engine the impulses occur at intervals of 120°, and in the case of an eight-cylinder V engine with the two sets of cylinders at 90° to one another, the firing impulses occur at intervals of 90°. In a twelve-cylinder engine with two rows or banks of six cylinders at 60° to one another, the firing impulses occur at 60° intervals.

Dorman Four-cylinder Engine. A transverse section through one cylinder of this engine is shown in Fig. 19. The bore of the cylinder is 69 mm. and the stroke 120 mm., while the b.h.p. developed at a speed of 2,000 r.p.m. is 25, and at a speed of 1,000 r.p.m. over 13. The engine as a whole forms a very stiff structure since the cylinder jackets and practically the whole of the crank chamber are cast in one from aluminium, the crank case not being divided on the centre line of the crank shaft. To the upper part of this casting is bolted a detachable cylinder head D, which forms the combustion chamber for all four cylinders and carries the valve guides and the pivots for the valve rocker arms. The lower part of the crank case is closed by a detachable aluminium casting C which forms the sump. The aluminium cylinder casting forms the water jackets for the cylinders proper which consist of cast-iron liners A. Each liner is forced into the aluminium casting from the top and a rubber ring is held under pressure between a shoulder turned on the outside of the liner towards its lower end and the aluminium casting. This ring makes a water-tight joint, and at the same time allows for unequal expansion. The inlet valves E and the exhaust valves F are operated by push rods J from cams on the two cam shafts K. When the detachable head D is removed it brings away with it all the valve gear, including the valve springs and rocker arms, the adjustment of which is not affected by such removal. The "breather" is led from the crank case up into the chamber formed by the top of the cylinder head and its cover O, so that while the engine is running an oily mist is continually rising from the crank case to the overhead valve mechanism ensuring continual lubrication. The crank shaft is supported by two bearings only, one of these being a very long plain bearing P at the fly-wheel end and the other a self-aligning ball bearing Q at the starting handle end. The plain bearing is mounted in a detachable end plate for the crank case, the opening being large enough to allow the crank shaft to be threaded through it in course of erection. The crank shaft is provided with balance weights. The trough system of lubrication is adopted in this engine for all bearings with the exception of the long crank shaft bearing, which is supplied under pressure from a pump in the sump which is rotated by a vertical spindle driven from one of the cam shafts.

Daimler Knight Sleeve-valve Engine. The essential and

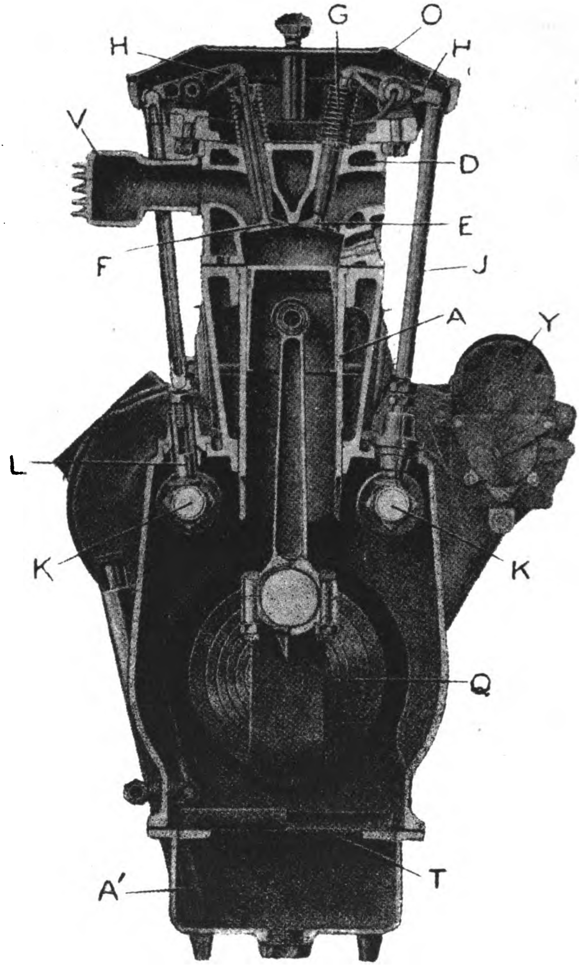


Fig. 19.

characteristic feature of this engine, of which a transverse section of one cylinder is given in Fig. 20, is the employment of two concentric sleeves A, B, reciprocated independently and interposed between the cylinder Q and the piston S. The use of two sleeves ensures rapid opening and closing of the exhaust and inlet ports, G, I, in conjunction with a sufficient area of opening.

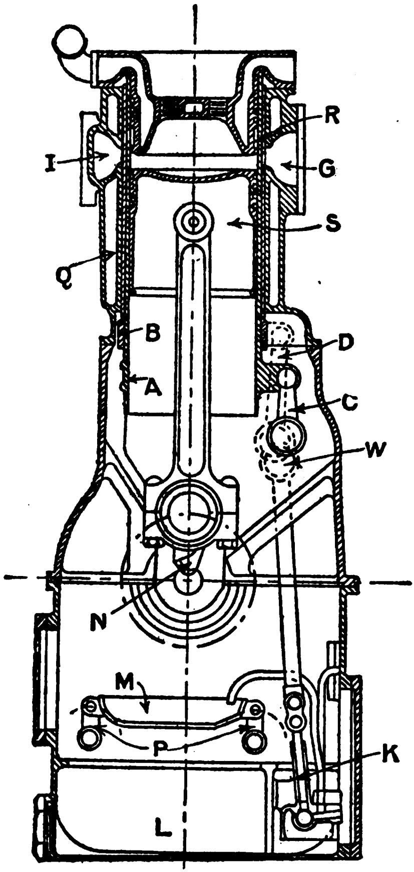
The inner sleeve A and the outer sleeve B are moved up and down by short connecting rods C, D, worked by the half-speed eccentric shaft W. These sleeves are of cast-iron and are quite thin. In a certain recent design of engine, having four cylinders 90 mm. diameter by 130 mm. stroke the thickness of the inner sleeve is 3½ mm. and that of the outer sleeve is 2½ mm. The stroke of each sleeve is 22 mm. The inlet port I on one side and exhaust port G on the other each extend nearly half way round the circumference of the cylinder and the sleeves are provided with corresponding ports.

The sleeves project upwards into a recess formed between the cylinder walls and the depending cylinder head. Leakage of gas is prevented by a broad junk ring R at the bottom of the cylinder head and three other rings above the junk ring.

The operating rods for the sleeves are placed at one side only, and obviously tend to tilt the sleeves. It is found in practice that no harm results from this arrangement in ordinary use. In one design of Knight engine for racing the sleeves were each operated by two connecting rods on opposite sides, two cam shafts being employed.

This sleeve-valve engine affords a good example of a lubricating system of the trough type. The oil pump K of the plunger type is worked from the eccentric shaft W. Oil drawn from the sump L is delivered to movable troughs M under the connecting rods, and also to fixed troughs (not shown) along the inside of the crank case, from which it flows to the five main crank shaft bearings. Dippers N on the ends of the connecting rods dip into the oil in the movable troughs M. The big ends are lubricated directly in this way, and the oil splashed up serves to lubricate the sleeves, etc. The movable troughs are carried by pivoted links P connected to the throttle control, so that the oil level is raised or lowered according to the opening of the throttle valve. In most engines with a lubricating system of this type, the troughs are not movable.

Aircraft Engines. During the war tremendous progress has been made in the development of the high speed petrol engine as applied to aircraft. The ratio of power to weight has materially increased, the consumption of fuel per horse power per hour has improved (this being of great importance in long-distance aeroplanes and on airships) and much higher powers have been obtained; such higher powers having been obtained by multiplication of cylinders, by the use of larger cylinders, and by an increase in the mean effective pressure. The development has in fact been characterised more by the attainment of higher mean effective pressure than by higher



A. Inner Sleeve.
B. Outer Sleeve.
C, D. Connecting-rods for Sleeves.
G. Exhaust Port.
I. Inlet Port.
K. Oil Pump.
L. Sump.
M. Oil Troughs.
N. Dippers on Big Ends.
P. Supporting Links for Oil Troughs.
Q. Cylinder Wall.
R. Packing Ring.
S. Piston.
W. Half - speed Eccentric Shaft.

Fig. 20.—Knight Sleeve Valve Engine.

speeds. Progress has been brought about by research and experiment, by improvement in design, and by the use of more suitable and stronger materials. For instance, cast-iron cylnders are now seldom used; the cylinders, water jackets, and combustion chambers are made of steel or the water jackets and combustion chambers are made of aluminium into which steel liners are compressed or shrunk. Pistons also are almost invariably of aluminium, aluminium alloy, or steel.

The Germans have used large six-cylinder engines developing up to 300 h.p. both on their Zeppelin airships and their night-bombing aeroplanes, and such engines have also been used—although to a somewhat limited extent—in this country. The majority of the high-powered aircraft engines made in England and America employ twelve cylinders in two "banks" or "rows" at an angle of 60° to one another and such engines have developed over 400 h.p. Still higher powers have been obtained by one or two engine makers by arranging three "banks" or "rows" of cylinders working on a common crank shaft, these rows being placed 40° apart.

Radial engines, both water and air cooled, have also made considerable headway, but the rotary air-cooled engines such as those of the Gnome type, have been largely superseded by fixed cylinder engines. Although the fixed cylinder type is in general heavier than the rotary type, its consumption of oil and fuel is so much less that the saving in weight of oil and fuel which has to be carried for a flight of longer duration than two or three hours more than compensates for the greater weight of the engine.

In the rotary engine the crankshaft is fixed to the fuselage or framework, and the crank case with its radiating cylinders rotates with the propeller. The Gnome engines are made with seven or nine cylinders arranged radially at equal angular intervals. When larger powers are required two seven or nine-cylinder sets are combined so as to form fourteen or eighteen-cylinder engines, the two cranks being close together, but at an angle of 180° to one another. These engines develop from 50 to 200 h.p., according to the number of cylinders.

In the radial engine, the cylinders are fixed and the crank shaft rotates with the propeller, but the arrangement of cylinders is similar to that described above in connection with the rotary engine. Such engines have been constructed to develop as much as 450 h.p. Both the radial and the rotary engines present certain difficulties in regard to valve operation, inlet and exhaust piping, and cooling, while both are responsible for greater head resistance than the "in line" engine and the rotary engine also wastes power by reason of the air resistance of the cylinders to rotation.

Sunbeam Aircraft Engine. A twelve-cylinder Sunbeam V engine giving 275 b.h.p. at a speed of 2,100 r.p.m. will now be described. A section through a pair of cylinders is shown in Fig. 21. The two rows or banks of cylinders are inclined at 60° to one another, and each cylinder has a bore of 110 mm.

and stroke of 135 mm. The cylinders are arranged in groups of three, and each group is provided with its own carburetter, the mixture from which passes through an induction pipe heated by a water jacket. A centrifugal pump at one end circulates the cooling water upwards through the cylinder and induction pipe jackets to the water outlet pipes arranged one above each cylinder. The six cylinders on each side each exhaust by a separate pipe into a main exhaust pipe of large diameter, both main exhaust pipes being arranged between the cylinders and leading into a single exhaust outlet.

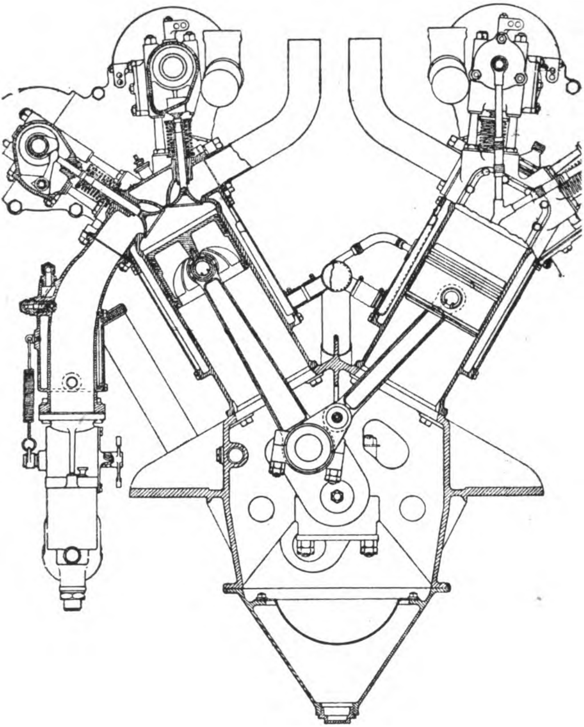


FIG. 21.

The crank shaft is carried in seven bearings. In many engines the propeller is connected to a short shaft arranged above the crank shaft and driven therefrom at a reduced speed by ordinary spur gearing. The gear ratio reduction varies but is generally about 50 per cent. In some engines the propeller shaft is arranged in line with the crank shaft and is driven therefrom at a reduced speed by epicyclic or other gearing.

In V engines in which two cylinders, one in each row, work on a single crank pin, several different methods of arranging the big end bearings are employed. In the Sunbeam engine one connecting rod works directly on the crank pin while the other is pivoted to a lug on the side of the first rod, so that it is always more or less in line with the crank pin. In some designs the big end of one of the connecting rods is forked, the big end of the other rod being disposed between the forks of the first rod, while in other designs the cylinders are placed slightly out of line and the big ends are arranged side by side on the same crank pin.

Two overhead cam shafts, driven through a train of gearing or a vertical shaft, run the whole length of each row of cylinders, one operating the inlet valves and the other the exhaust valves. Each cylinder is provided with two inlet valves and two exhaust valves, these valves being operated by cams acting on the valve stems through the specially shaped end of a pivoted lever. The arrangement of valves in tho cylinder head as shown clearly makes the combustion chamber of very simple and regular shape and so facilitates rapid combustion, while the employment of multiple exhaust and inlet valves assists in the rapid transfer of the mixture and the exhaust gases to and from the cylinders, this being a feature of considerable importance in high-speed engines.

Two independent twelve-cylinder magnetos are used, the connections from one magneto leading to one set of sparking plugs while the other magneto connects to an independent set of plugs. The order in which the cylinders fire is 1, 1a, 5, 5a, 3, 3a, 6, 6a, 2, 2a, 4, 4a.

All "in line" engines are lubricated on the dry sump system, this being essential in view of the fact that such engines have to work at considerable inclination, while the oil also keeps cooler. In the Sunbeam engine the sump tapers comparatively sharply to a point in which a drain plug is fitted. The lubrication system is known as "compound pressure" and is similar to the pressure system described in connection with the Star engine, but the oil is delivered at a much higher pressure in view of the very severe working conditions. A duplex gear wheel lubricating pump placed at the lowest part of the sump is driven continuously from the crank shaft. One part of this pump draws oil from a tank and forces it under pressure to the cam shaft and big end bearings. The surplus oil which collects in the sump is returned by the other part of the pump to the oil tank, so that it is not exposed to the heat of the crank case for any length of time.

A pressure gauge is connected to the oil delivery pipe and the oil pressure should be maintained considerably above 20 lbs. per square inch. When the oil pressure falls below this limit or when the engine speed exceeds 2,500 r.p.m. a governor cuts out the ignition. Such a pressure can only be maintained if the oil is prevented from becoming overheated (i.e. above about 60° Centigrade). An oil cooler may be necessary.

A larger Sunbeam engine than that described above having cylinders of 160 mm. bore and 169 mm. stroke develops 400 h.p. at 2,000 r.p.m. The weight of this engine complete but without water, fuel, or oil is 1,000 lbs., while the gross weight of the engine in running order with fuel and oil for 6 hours is 2,519 lbs. The oil consumption is ·032 pints per b.h.p. per hour, and the fuel consumption is ·58 pints per b.h.p. hour.

CARBURATION.

Surface carburetters are now practically obsolete, the jet type being universal. A spray of petrol is projected from a small jet in the current of air flowing towards the engine, and a great variety of constructions is in use, all endeavouring to proportion in an efficient manner the supply of fuel to the varying requirements of the engine.

A motor-car engine is required to run at speeds varying from a few hundred to some thousands of revolutions per minute, to exert a large or small torque or turning effort at either high or low speeds, or even to turn slowly and quietly without giving out any useful power: also to accelerate rapidly and start easily when cold without special preparation. A carburetter should further give economical running, should be simple and cheap, and should not require skilled adjustment. A carburetter which will automatically adjust itself to these extremely varied conditions requires to be very carefully designed. That the problem has not yet been finally solved is shown by the great variety of carburetters in use.

The simplest form of "jet-in-tube" carburetter, which would be satisfactory when applied to a stationary engine giving approximately constant power, suffers when applied to motor vehicles from the defect that the mixture becomes richer the higher the engine speed, producing an excess of petrol at high speeds and a deficiency at low speeds. In some carburetters this tendency is neutralized by fitting additional jets and making the flow through these jets independent of the suction of the engine. In other carburetters the ratio $\frac{\text{petrol}}{\text{air}}$ is kept constant at all speeds by regulating the size of the petrol opening or the air inlet or both, either by a connection to the throttle valve or by means of an independent moving part.

Zenith Carburetter. In principle this carburetter, Fig. 22, consists of a simple jet-in-tube carburetter (central jet G in choke tube or main air supply passage X) with the addition of a compensating jet H (annular) and a starting and slow-running jet *a*. The main jet G is supplied with petrol from the constant level float chamber, and the flow depends upon the suction at the point S (which again depends on the opening of the throttle valve P) and the size of the main jet at S. The flow through the annular compensating jet is not affected by the suction at S but depends upon the size of the jet I and the suction in the passage F. The small air opening O at the top of the vertical tube above the jet I insures that the suction in

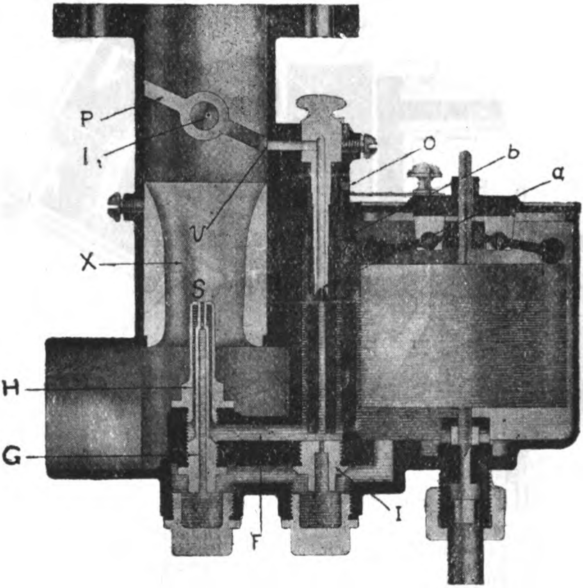


FIG. 22.

the passage F is practically constant and independent of the suction at S. At high speeds the deficient supply through the annular jet H compensates for the excess through the central jet G. During ordinary running the petrol level in the vertical tube above the jet I is low, or the tube may even be empty, but when the engine slows down the level rises nearly or quite to the level in the float chamber and forms a reserve which can be drawn quickly into the cylinder to give rapid acceleraton when the throttle P is opened. The starting and slow-running jet *a* is subject to suction through the passage U, which opens against the edge of the nearly closed throttle valve P. A small quantity of air from the opening O is then also drawn through the holes *b* and past the jet *a*.

S. U. Carburetter. In this carburetter, shown in section in Fig. 23, the size of the petrol jet, the size of the air passage, and the suction on the jet, are automatically varied by a movable piston or plunger A. A tapered needle valve B fixed to and concentric with the piston A passes into the jet C. As the piston and needle move in or out the effective size of the jet is made smaller or larger, and the size of the choke tube R is also varied in such a way as to give the correct mixture. The upper end of the piston rod carries a mushroom disc, forming the movable end of a bellows chamber E, connected by a passage F to a point in the air conduit near the usual throttle valve D.

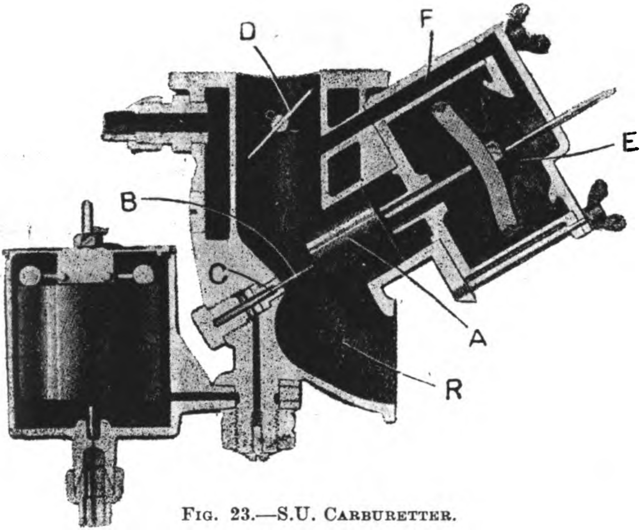


FIG. 23.—S.U. CARBURETTER.

- **A.** Movable Piston.
- **B.** Tapered Needle Valve.
- **C.** Jet.
- **D.** Throttle Valve.
- **E.** Bellows Chamber.
- **F.** Passage to Bellows Chamber.
- **R.** Choke Tube.

The operation of the bellows and the consequent regulation of the petrol and air supply thus automatically depends upon the suction of the engine.

The latest S.U. carburetters are provided with what amounts to a small auxiliary carburetter, shown in Fig. 24, by means of which a rich mixture may be obtained when starting or when it is desired that the engine should exert maximum torque at low speeds while still leaving the main carburetter tuned for economical running. A small rotary valve G operated independently puts the passage H into communication with the part of the induction pipe above the throttle valve, so that a small quantity of air is drawn through the opening J and carries with it up the passage H a quantity of petrol admitted through the orifice L.

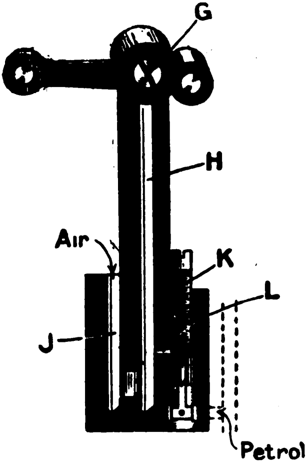


FIG. 24.

Solex Carburetter. This carburetter shown in Fig. 25 is extensively used, particularly on commercial vehicles and buses. The float F acts directly on the needle valve *p* so as to keep the petrol level constant. The movement of the rotary throttle valve V is limited by a screw Z, this screw being adjusted so as to position the valve for slow running when air and petrol are drawn through the auxiliary jet channel *y*. The main jet is specially designed to correct the tendency, referred to previously, of a simple jet-in-tube carburetter to deliver too rich a mixture at high speeds. The main jet G communicates through a small gauged opening in its lower end with the float chamber and is arranged concentrically within an annular upward extension of the jet stand *t*. The jet cap A when screwed down holds the jet in position. Holes in the jet cap and jet are arranged so that air may enter at the lower end of the jet cap, may pass over the top of the annular extension of the jet cap *t*, and then enter at the bottom of the jet. The space available for the main air supply between the jet and choke tube K is restricted so that the pressure at this point decreases rapidly as the engine speeds up. This reduction of pressure does not, however, cause an excessive suction on the jet, since the air admitted through the holes at the lower end of the jet tube G prevents a correspondingly great reduction of pressure at this point.

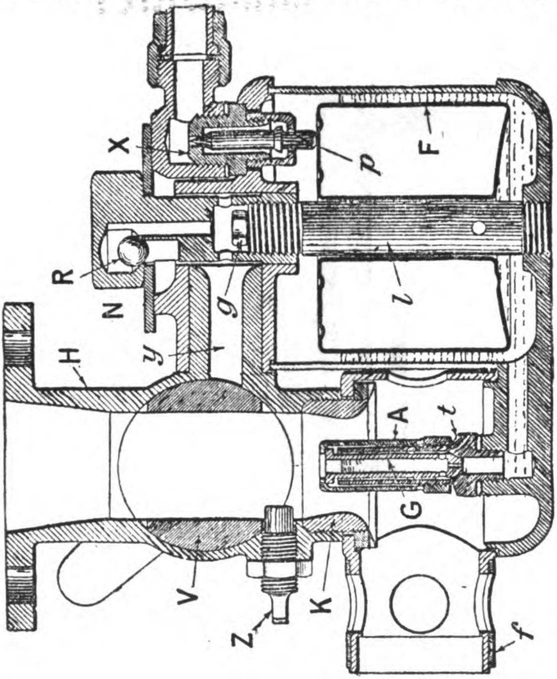


FIG. 25.

During slow running a reserve of petrol accumulates in the interior of the main jet G and is thus immediately available when the throttle is opened and rapid acceleration is desired. When the throttle valve is turned clockwise for starting or slow running as far as the adjustable stop Z allows, the main opening is nearly closed and the auxiliary jet channel *y* communicates with the throttle chamber H and induction pipe. Petrol is then drawn through the auxiliary jet *g* and its hollow stand *l*, while air is drawn past the ball valve R. When the nut N is unscrewed the float chamber and the main and auxiliary jets may be removed so that adjustments may be readily made.

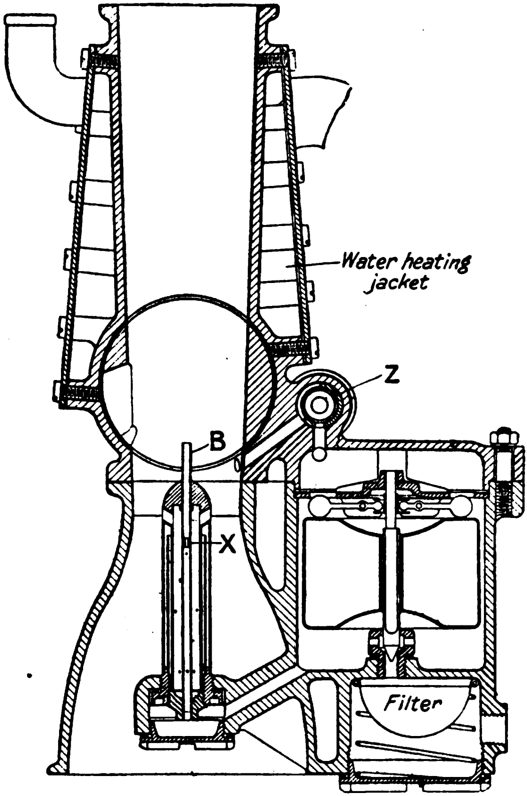


FIG. 26.

Claudel-Hobson Aircraft Carburetter. In this carburetter, shown in section in Fig. 26, as modified to suit the Napier aero engines, the float and float chamber are of normal design, and the feature of primary interest is the diffuser or jet assemblage, which is shown separately in Fig. 27. *When the throttle valve is in the slow-running position, a narrow slot in its edge passes over the slow-running tube B, upon which the engine* suction is then concentrated. Petrol is drawn through the restriction at X forming the pilot jet and air through the holes E, so as to give a correct slow-running mixture. Petrol then enters the diffuser through the main jet A at a greater rate than the pilot jet X can dispose of, so that a reserve accumulates in the diffuser tube N and guide tube M, and blankets the air holes F in the diffuser tube. As the throttle opens, the suction is communicated to the diffuser as a whole, air is drawn into the diffuser through the holes H, and a mingled stream of petrol and air passes out through the emulsion holes G, where it meets the main air stream passing up the choke tube. The petrol level in the diffuser then falls and the air holes F are progressively uncovered thus relieving the main jet A of a portion of the suction and automatically counteracting the tendency towards an unduly rich mixture at high speeds. This carburetter is fitted with an altitude control by means of which the tendency at high altitudes and correspondingly low atmospheric pressures towards an unduly rich mixture is counteracted by reducing the rate of flow of petrol. This is effected by making the upper part of the float chamber air tight and regulating the pressure on the upper surface of the petrol.

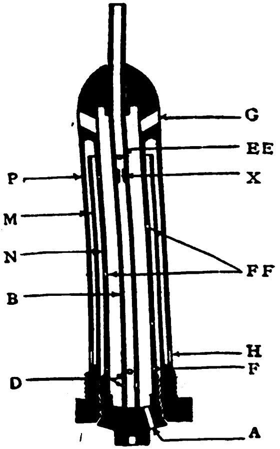


FIG. 27.

At lower altitudes the altitude control valve puts the upper part of the float chamber into communication with the atmosphere, but at higher altitudes the valve opens a passage between the float chamber and choke tube so that the suction of the engine is communicated to the upper surface of the petrol and reduces the rate of flow through the main jet. The amount the valve is opened depends upon the altitude.

S. U. Cycle Engine Carburetter. The carburetters described previously have been intended for multi-cylinder engines of the car or aircraft type. Smaller and simpler carburetters are required for small single and two-cylinder motor cycle engines. Many of the carburetters in use have in addition to the main throttle lever a second lever for an additional air valve, but there is a tendency towards the increasing use of what is called the single-lever carburetter. The S.U. carburetter is an example of this type, a section being shown in Fig. 28, and an external view at right angles thereto in Fig. 29. This carburetter is what is known as the straight-through pattern; that is, air admitted through the gauze covered air intake passes directly and in an horizontal direction to the outlet B, which is bolted

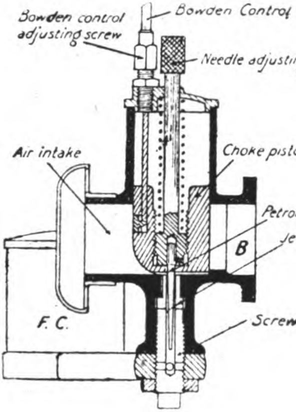


FIG. 28.

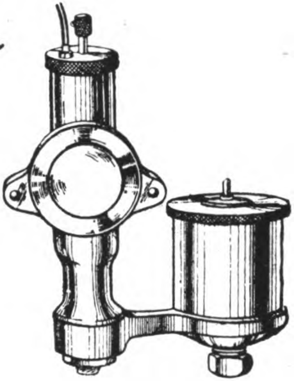


FIG. 29.

direct to the engine or to a short induction pipe. In this passage is a piston which forms the throttle valve and is moved up and down by means of a Bowden wire connected to a lever on the handle bars. The movement of the piston also varies the effective size of the jet by means of a taper needle which is screwed into the piston and enters the jet so that when it is moved up and down with the piston, the effective area of the jet is increased or decreased as the throttle opening increases or decreases. In the closed position the petrol jet is completely closed and the air is also shut off. The taper needle really co-operates with the jet as in the large S.U. carburetter so as to supply petrol and air in more or less correct proportions, but it is moved directly by the throttle piston instead of by the plunger and bellows. The plunger is pressed down into its closed position by a helical spring and means are provided for adjusting the taper petrol control needle from the outside of the carburetter, consisting of a rod which can be screwed into and out of the piston by its knurled end to vary the proportions of air and petrol. The carburetter may be so placed on a motor cycle that the driver can adjust the needle from the saddle. A carburetter of this type will on a machine of about 2½ nominal h.p. give a consumption of more than 90 miles per gallon, the weight of the rider and machine being over 300 lbs.

Paraffin Vaporizers. Horizontal engines in which paraffin vaporizes form part of the engine and ignition is effected by a hot chamber are described later. Special paraffin vaporizers constructed in general as attachments to the engine and not designed as part of the engine itself have been used in multi-cylinder vertical engines of the motor car type applied to lorries, agricultural tractors, and boats. A very large amount of heat is required to vaporize paraffin satisfactorily and vaporizers consist generally of an attachment containing adjacent passages for the exhaust and for the paraffin mixture. The Thornycroft vaporizer consists broadly of a cylinder arranged across one end of the engine, the exhaust pipe leading into one end of this cylinder, from the exhaust side of the engine, while the paraffin mixture is led away from the other end to the inlet valve. Figs. 30, 31, and 32 show diagrammatically the essentials of the vaporizer.

Air enters the end of the pipe B and paraffin is drawn as required into this current of air through a jet which is just inside the entrance to the pipe B. The mixture then passes along the pipe B and U-tube H in the exhaust jacket A. A special feature of this vaporizer is the means by which the heating effect of the exhaust gases is regulated. The bye-pass valve F, of special shape, co-operates with a special construction of exhaust jacket or vaporizing chamber, so as to pass more or less of the exhaust gases round the pipes H through which the paraffin mixture travels. The vaporizer is constructed of an outer cylindrical casing J and a horizontal diaphragm K, which extends from the valve F nearly to the other end of the vaporizing cylinder A. When the valve F is in its horizontal position its edge coincides with the diaphragm K, and all the exhaust gas is consequently compelled to pass right along the upper side of the cylindrical vaporizer, downwards past the end of the diaphragm K and back along the lower side of the vaporizer. When the valve F is turned into a vertical position the exhaust gases flow right through and the heating effect of the vaporizer is very small. By adjusting the valve F any desired heating effect is obtained. The valve F does not require frequent adjustment; it can be set in accordance with the prevailing atmospheric temperature and will only require alteration if the temperature varies substantially or the quality of paraffin alters. For the ordinary English summer climate about half the exhaust gives sufficient heat for running, but it is to be noted that it is desirable to have too much heat rather than too little. With all engines having paraffin vaporizers, it is necessary to start the engine on petrol. Paraffin or petrol can be turned on at will.

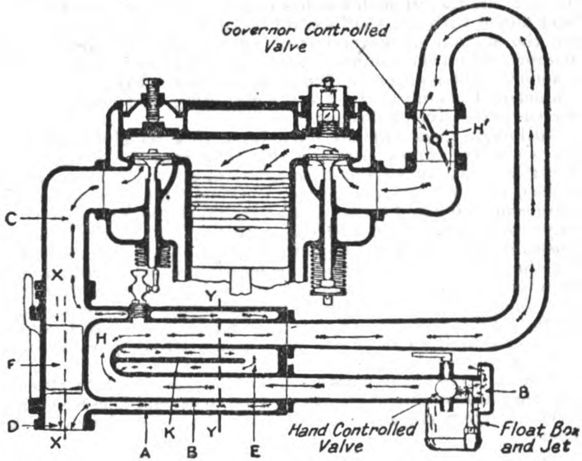
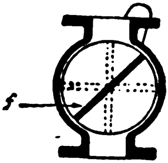
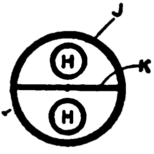


FIG. 30.



Section X X
FIG. 31.



Section Y.Y.
FIG. 32.

OIL ENGINES.

Under this heading are considered engines using (1) kerosene or paraffin; (2) crude or unrefined oils; or (3) residual oils, that is, oils from which the more volatile constituents have been removed. Oils of this character, particularly those mentioned under (2) and (3), require a considerable amount of heat to vaporize them sufficiently to form an explosive mixture.

Oil engines for use with paraffin have been made for many years; the fuel in such engines is generally vaporized by heat and mixed with air, the mixture being drawn into the cylinder on the suction stroke and then compressed; or air alone is compressed on the compression stroke and then mixed with the vapour. The ignition is effected by the heat of the walls of an unjacketed chamber or by some refractory material kept hot by the successive explosions.

Although these engines have been very widely used they have always been open to the objections that the compression must be kept low to avoid pre-ignition, and that more or less refined oil is always required for working, while in many countries crude oil could be obtained very much more cheaply than any other fuel. Such fuel can only be used satisfactorily in Diesel engines or in the modern hot-bulb engine, in which the oil is injected by a force pump, or by compressed air into a bulb or unjacketed chamber kept hot by the heat resulting from the previous explosion and from the compression. The temperature of the bulb is sufficient to vaporize and ignite the fuel, which is injected at the end of the compression stroke, pure air only being compressed.

Some engines of this class resemble Diesel engines very closely, and they are sometimes loosely referred to as semi-Diesel engines.

Oil-injection engines work on both the two-and four-stroke cycles.

Ruston-Proctor Paraffin Engine. These four-stroke engines are of the kind in which a mixture of air and vapour is compressed. They are made in various sizes, developing from 3 to 55 maximum h.p. The continuous working load for periods exceeding two consecutive hours should be about 10 per cent less than the maximum load. These engines are in their general characteristics somewhat similar to the horizontal gas engines described previously.

The clearance space, Fig. 33, is very large and the compression is, in consequence, low, being only about 50 lbs. per square inch. The pure air supply is drawn past a throttle valve H, which may be set by hand, and then through the upper valve G. A rotary valve F, operated from the cam shaft, connects the cylinder, when required, with the vaporizer A and the ignitor B, which are not water-jacketed, and remain at sufficiently high temperatures to effect respectively the vaporization and ignition of the fuel. A small oil pump D feeds the oil from a tank to the vaporizer A by means of the pipe J.

The vaporizer A and the igniter B are heated by a lamp before starting. The engine is then started by giving a few turns to the fly-wheel, the compression releasing device being then in operation.

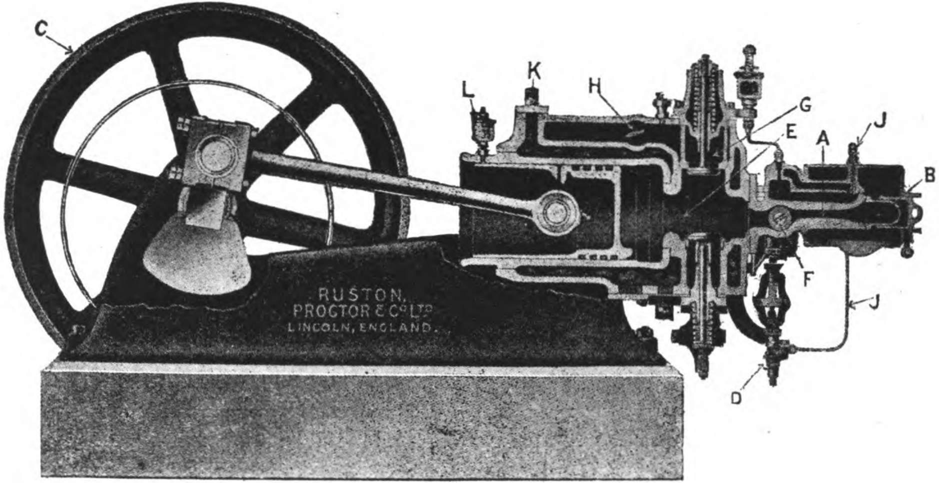


FIG. 33.—RUSTON-PROCTOR FOUR-STROKE PETROLEUM ENGINE.

A. Vaporizer.
B. Igniter.
C. Flywheel.
D. Fuel Pump.
E. Combustion Chamber.
F. Ignition Valve.
G. Main Air Valve.
H. Air Throttle Valve.
J. Fuel Pipe to Vaporizer.
K. Cooling Water Outflow.
L. Lubricator for Piston and Gudgeon Pin.

During the suction stroke the vaporized oil is drawn into the cylinder through the then open vapour valve F, and air is simultaneously drawn in through the main air valve G.

The valve F is closed during the compression stroke, but is opened at the end thereof, and the explosive mixture is ignited, in consequence of the high temperature of the vaporizer and igniter.

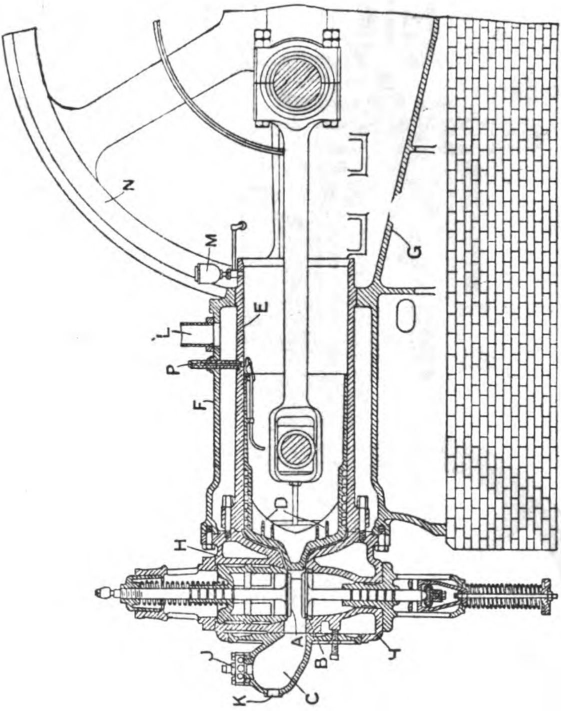


FIG. 34.

When it is necessary to run at exceptionally light loads the heat of explosion may not be sufficient to keep the vaporizer hot enough to ignite the mixture. In such a case it may be occasionally re-heated, or a small lamp may be left burning continuously.

Ruston-Proctor Crude Oil Engines. Vaporizers as described above can only be used with continued success on engines burning paraffin. The result of attempting to use a vaporizer with crude or residual oils is merely to distil or vaporize the lighter or more volatile constituents for use, leaving behind a carbon deposit which necessitates frequent stopping and cleaning of the engine. This objection is largely overcome in engines of the Diesel type and hot-bulb engines by injecting the fuel directly into the cylinder at the end of the compression stroke and compressing only pure air. This results in rapid combustion of both the light and heavy constituents of the oil, and avoids heavy carbon deposits in the combustion chamber and on the valves, while the greater compression possible gives higher thermal efficiency.

In the Ruston-Proctor four-stroke engine, Fig. 34, the pressure at the end of the compression stroke is about 300 lbs. per square inch. The upper valve A admits pure air and the lower valve B serves for exhaust as before, but the two are closer together, so as to reduce the clearance volume. With the same object the piston is shaped so as to approach closely to the end of the cylinder. The fuel oil is injected at the end of the compression stroke into a hot bulb C *not water-jacketed*, the bulb forming part of the clearance volume.

The piston is very long, and cooling ribs D are cast on the inside. To the outer end of the combustion head is bolted the flange of the hot bulb C. The fuel injection valve J is bolted to the top of the bulb. To prevent the hot bulb from becoming too hot a water-snifting valve is arranged horizontally in the end K of the bulb. During the suction stroke this valve admits a small quantity of water. A cock is fitted to regulate the water feed to the snifting valve, so that it keeps the bulb externally at a black heat, considerably below red heat.

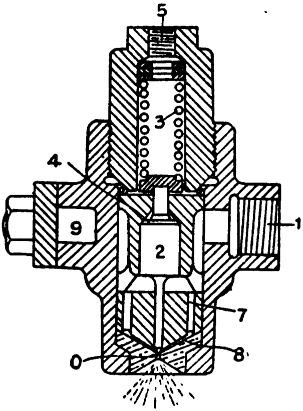


FIG. 35.

In hot bulb engines working with a high compression pressure, as in the engine now under consideration, the injection of the fuel into the hot bulb presents some difficulty. Some engines, e.g. the Blackstone engine, to be described later, use compressed air, but most, e.g. the Ruston—employ a mechanical injector or atomizer, shown in Fig. 35. At or about the end of the compression stroke a small quantity of fuel is forced to the atomizer, through a pipe connected to the opening 1. A duplex valve 2, loaded by a powerful spring 3, is lifted by the pressure of oil at each stroke of the oil fuel pump; the smaller valve seat closes the opening to the combustion chamber, while the larger seat 4, when the valve is in its raised position, closes a leak pipe 5 connected to the suction side of the oil pump. The pressure of the oil acting on the lower end of the large part of the valve lifts it and forces it up to its larger seat 4. When the valve lifts, the oil is forced through passages 7 and spiral channels 8, converging to a central outlet O, so that it emerges on the other side of the nozzle in a cone-shaped cloud of oil particles. The movement of the valve between the limits set by its larger seats is about $\frac{1}{10}$ inch. Some of the ordinary cooling water is circulated through a passage 9 partly surrounding the valve body.

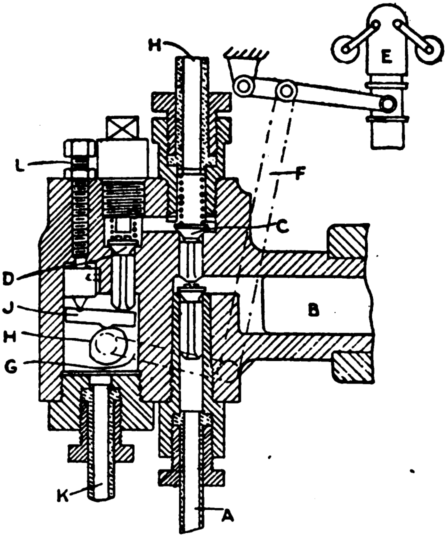


Fig. 36.

A diagrammatic sectional view of the fuel oil pump and governing gear is shown in Fig. 36. A special cam on the end of the cam shaft operates the plunger B on the pumping stroke. The cam is designed to give a very rapid delivery stroke, to assist in reducing the oil to a very fine spray. The fuel is drawn from the oil tank through the suction pipe A, and is delivered through the pipe H. A full charge of oil is pumped past the delivery valve C on each stroke, but some is returned to the fuel tank through an overflow valve D, the amount so returned being regulated directly by the governor acting through a cam H, which varies the lift of the valve D according to requirements.

Blackstone Crude Oil Engine. This four-stroke engine possesses two distinctive features—

(*a*) The fuel is injected by means of compressed air, as in the Diesel engine, instead of a mechanical atomizer.

(*b*) Only a small portion of the fuel is injected into the hot bulb proper, the greater part being injected directly into the cylinder.

Compressed-air injectors are better than mechanical injectors for effecting atomization, although this advantage is to some extent neutralized by the added complication of an air compressor. When hot-bulb engines are running light the heat of the burning gases is often insufficient to keep the bulb hot enough to ignite the fuel, and it may be necessary to heat the bulb from the outside as at starting.

In the Blackstone engine this difficulty is avoided by the dual injection system. A practically constant quantity of fuel is injected into the hot bulb whatever the power developed, the amount of fuel sprayed direct into the cylinder being varied according to requirements. A rod from the governor, which is driven direct from the crank shaft, acts on a wedge which regulates the stroke of the fuel pump, the fuel being led to the dual injection devices at the other end of the engine. A two-stage air compressor, arranged at the side of the working cylinder and driven by an eccentric, is mounted at the side of the engine.

In Fig. 37 is shown a sectional elevation looking at the end of the cylinder and showing the end A of the combustion chamber, the hot bulb B, and the fuel injection box C. In Fig. 38 is shown a sectional plan on the line XX.

Two needle valves, D, E, are operated together and are normally kept on their seats by springs F, G. The injection box C is kept filled with compressed air and the fuel is pumped into an annular space H surrounding the valve D which controls the injection into the hot bulb. The overflow from this space passes to a similar annular space P surrounding the main injection valve E, which is placed at right angles to and slightly below the valve D. At the end of the compression stroke both valves are slightly lifted, and some air is forced past, carrying the fuel with it. As the pressure of air in the injection box is about 400 lbs. per square inch, and the compression pressure is only about 150 lbs., there is clearly an ample margin of pressure to ensure the injection of the air and fuel at a very high velocity.

Both valves are simultaneously raised from their seats by a vertical spindle J operated from the cam shaft and carrying arms L, M. In later designs the operating levers L, M are tripped or released suddenly so that the valves are returned sharply to their seats by the closing springs F, G.

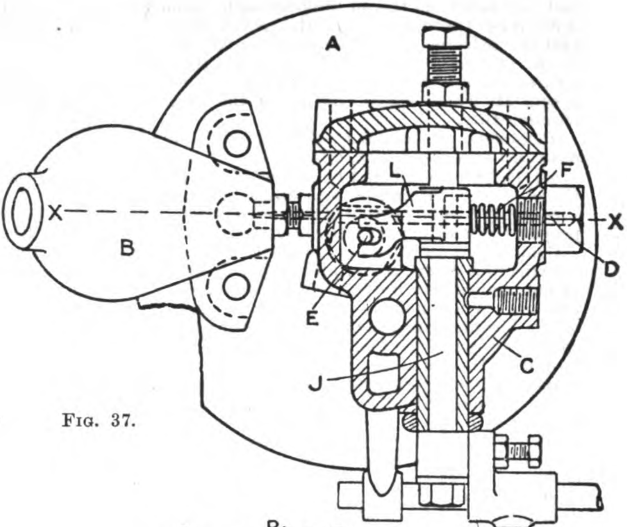


FIG. 37.

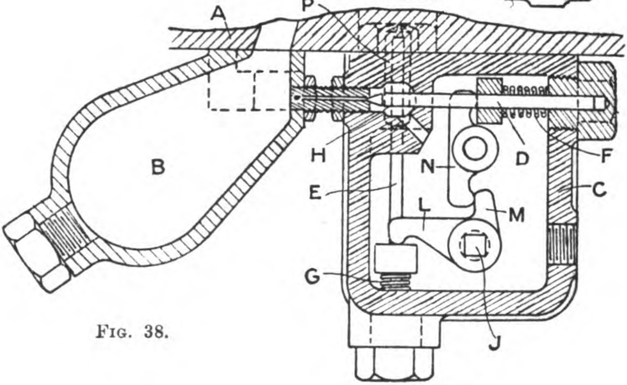


FIG. 38.

The ignition spray may be advanced or retarded relatively to the main spray by adjusting the position of the pivot of the

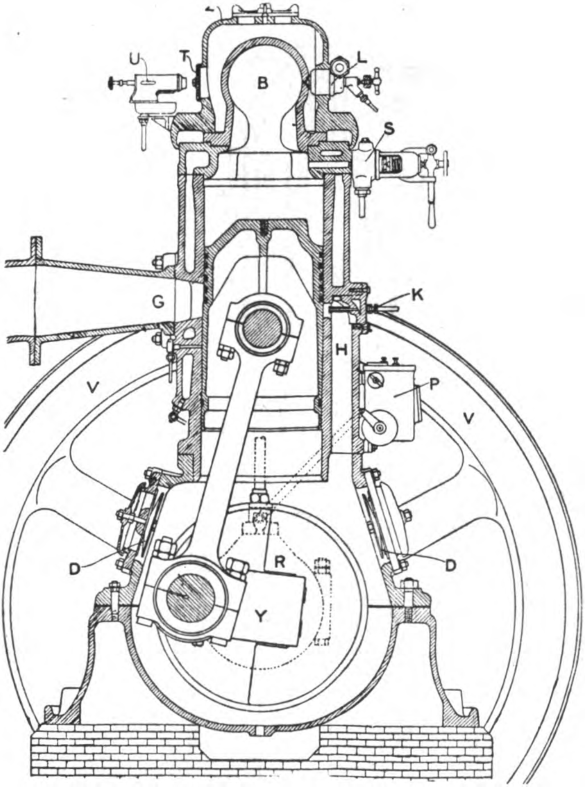


FIG. 39.

rocking lever N, the pivot being for this purpose carried eccentrically on a rotatable member which can be adjusted while the engine is running.

Robey Crude Oil Vertical Engine. This two-stroke engine, shown in Fig. 39, is made in several sizes developing from 20 to 30 b.h.p., while a smaller design develops from 4 to 20 b.h.p. The crank case is used as the air pump, air being discharged through the transfer passage H at a pressure of about 8 lbs. per square inch. The cylinder is water-jacketed, but the hot

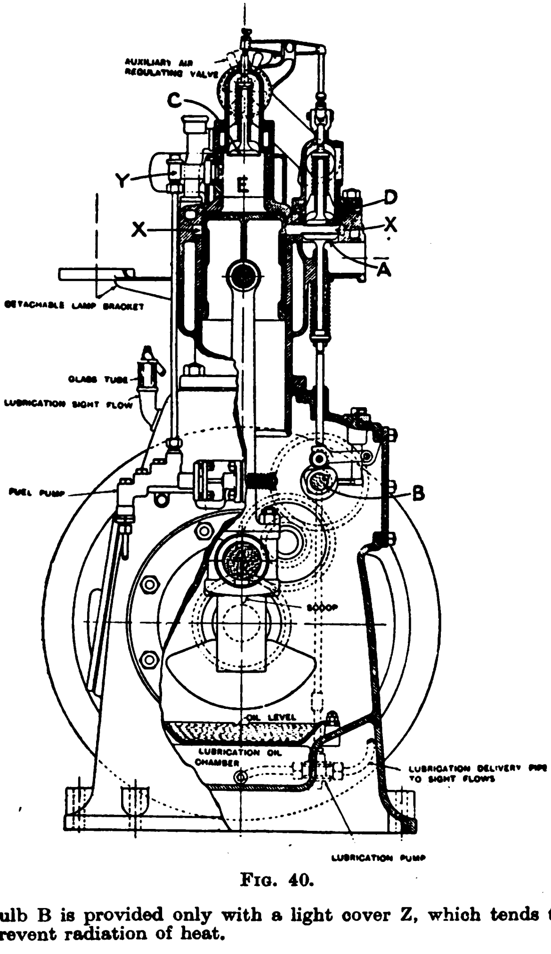


FIG. 40.

bulb B is provided only with a light cover Z, which tends to prevent radiation of heat.

The fuel is injected direct into the hot bulb through the valve L, designed to give a whirling motion thereto. The fuel pump and the lubricating oil pump are both worked from an eccentric R on the crank shaft, the eccentricity being varied by the governor. Overheating is prevented at full loads by the injection of a small quantity of water through the valve K by means of the cooling water circulating pump, the amount being regulated automatically or by hand. A removable cover T allows the operation of the starting lamp U.

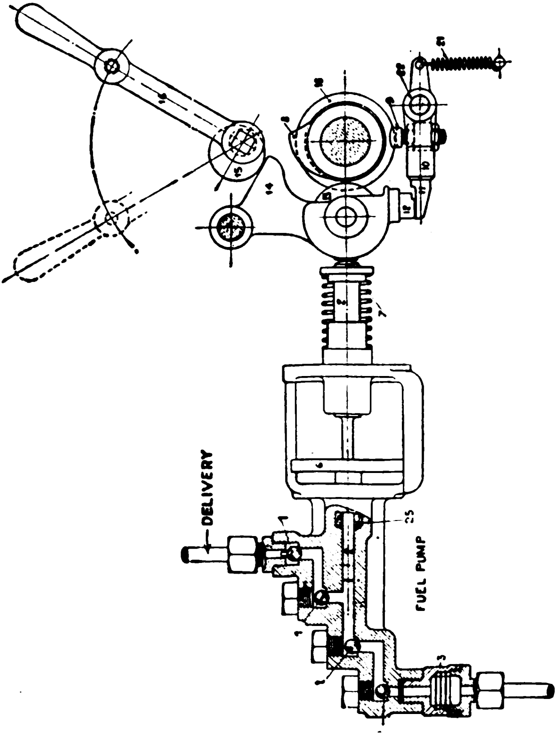


Fig. 41.—Fuel Pump and Actuating Gear.

Cross Vertical Engines. This four-stroke engine manufactured by the Westinghouse Brake Co., Ltd., possesses several distinctive features. Fig. 40 shows a section of a 10 h.p. single-cylinder type. The exhaust valve A is operated directly from the half-time cam shaft B and the two air inlet valves C, D are also operated from the cam shaft but through rocking levers. The inlet valve C opens directly into the combustion chamber E; the other inlet valve D and the exhaust valve A open into an annular pocket or chamber X, which is cut off by the piston when the latter is in its highest position. The combustion chamber E is unjacketed and is thus equivalent to a hot bulb, but the inlet valve C at the top of the combustion chamber is placed on a water-jacketed seating. When starting, a small iron bulb (not shown), screwed into the combustion chamber E, is heated by a lamp and serves to ignite the charge for the first few revolutions. During the suction stroke air is drawn into the cylinder through the valves C and D. On the compression stroke the piston first compresses the whole of the air in the cylinder. Towards the end of the stroke the annular chamber or pocket X is covered by the piston and the air trapped in the combustion chamber is further compressed to a pressure of about 200 lbs. per square inch. The oil fuel is now sprayed through the injection valve Y into the combustion chamber, the air in which is only sufficient to burn a small quantity of the fuel. The combustion is thus incomplete until the movement of the piston downwards uncovers the annular chamber X, which contains heated compressed air and so completes the combustion during the further movement of the piston.

Combustion is thus spread over a considerable movement of the piston and the rise of pressure when the fuel is admitted is not as sudden and as high as usual. The fuel pump is shown in section in Fig. 41. It is controlled by the governor so as to give either a long or a short stroke according to the load; oil is thus always injected and no power strokes are missed entirely. Two suction and two delivery ball valves 1 are employed and the oil is strained by passing it through layers of fine gauze 3 before it is admitted to the pump. The plunger 2 is moved to the left for the working stroke by means of a cam 8 on the half-speed shaft bearing against a roller 13 carried by a lever 14. The return stroke is brought about by a spring 7. When it is necessary to reduce the power of the engine the forward stroke of the pump is reduced by limiting the return movement of the arm 14. This is effected (1) by the hand lever 16 and eccentric 15 or (2) by the governor raising the arm 10 so that it catches the end 12 of the arm 14.

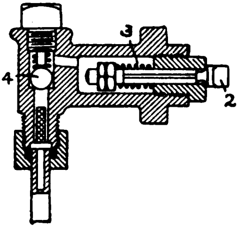


Fig. 42.—Fuel Injection Valve.

The automatic fuel injection valve Y, which is screwed into the side of the combustion chamber, is shown in section in Fig. 42.

PRODUCER GAS PLANT.

Gas Production. Producer gas is generally understood to mean gas obtained by partially burning fuel in a special gas producer. It is distinguished by this partial combustion from coal gas, which is produced in closed retorts heated by an external and entirely independent source of heat, whereas in a gas-producer all the heat required is obtained from the fuel from which the gas is obtained. The gas produced consists generally of hydrogen (H), carbon monoxide (CO), hydrocarbons (such as CH_4), carbon dioxide (CO_2), and nitrogen (N), mixed together in various proportions.

Practically any kind of fuel containing carbon can be used in gas producers, although some fuels are much more difficult to deal with than others. Anthracite coal and coke are the simplest fuels, and are generally used in small installations, to escape the difficulty which arises when tar is one of the products ; but other fuels, such as bituminous coal, peat, sawdust, or waste wood, can be used, generally in specially designed producers.

Water Gas is made by passing steam and air alternately through carbon, the alternations taking place at intervals of a few minutes. The air is blown through with the object of maintaining the carbon in a state of incandescence. The principal constituents are hydrogen (H) and carbon monoxide (CO). Its calorific value is about 300 B.Th.U. per cubic foot.

Steam as the result of its action upon incandescent carbon is split up into its two constituents, hydrogen and oxygen. This process requires the application of heat and the action may be represented by the equation—

$$H_2O + 62{,}000 \text{ B.Th.U.} = 2\,H + O.$$

Simple Producer Gas, or air gas, is obtained by blowing air through a deep bed of coke or other carbon fuel, part of which is incandescent. The fuel is only partially burnt, owing to the supply of air being strictly limited and insufficient for complete combustion. The principal constituent is carbon monoxide (CO) if coke is used, but if the gas is made from coal there is also some hydrogen (H) and methane (CH_4), which gives a higher calorific value. About 30 per cent of the heat of combustion of the fuel is generated in the producer, the remaining 70 per cent being available for subsequent use. The calorific value is from 70 to 100 B.Th.U. per cubic foot.

The principal action in the producer may be represented by the equation—

$$C + O = CO + 4{,}400 \text{ B.Th.U.}$$

Whether this action really occurs directly or whether there is an intermediate formation of carbon dioxide, is uncertain. It is possible that both actions take place.

Mixed Producer Gas, or Dowson gas, is intermediate between water gas and simple air gas. A certain proportion of steam is mixed with the air which passes through the incandescent fuel.

In a mixed producer gas plant both of the above actions occur simultaneously, each to such an extent that the heat resulting from one action to a great extent balances the heat required for the other, so that no external heat is required and no cooling is necessary.

Suction Gas Plants consist of a gas engine combined with a gas producer in such a way that the amount of gas generated depends upon the suction of the engine, so that the gas is generated solely according to requirements. Plants of the suction type are in very common use, and are made in sizes intended to develop from about 50 to 300 horse-power. Practically every gas-engine maker is prepared to supply a suction gas plant for use with his engine.

Mond Gas, named after its originator, is produced from bituminous coal on a large scale, and in a very economical manner, owing (1) to the means for recovering ammonia, and (2) to the means for utilizing the heat inseparable from any gas producer. The gas consists largely of hydrogen (H) and carbon monoxide (CO), with a small quantity of methane (CH_4). The calorific value is about 150 B.Th.U. per cubic foot. The very low calorific values of gases generated wholly or in part by air are accounted for by the very large proportion of nitrogen in the air supplied.

Poor Gas is a term applied somewhat vaguely to producer gas owing to its low calorific value.

National Suction Gas Plant. This apparatus is designed to use anthracite coal, gas coke, or charcoal, and it consists of the generator or furnace A, in which the fuel is decomposed, and the scrubber B, in which the gas evolved is cooled and cleaned. The engine draws its charge of gas from the expansion box I, Fig. 43, which is directly connected with the top of the scrubber by the gas main 2. The scrubber in turn receives its supply of gas from the gas outlet pipes, which communicates with the furnace 4 of the producer.

Every time the engine sucks in a charge of gas the suction action is communicated from the engine through all the interior connections of the gas plant until it is felt at the furnace of the producer, and consequently the production of gas is quite automatic and in accordance with the demand made by the engine.

The gas which comes off at the upper part of the producer is used to vaporize the water required for the steam supply and to heat the air entering at inlet 6. A jacket 5 through which passes the incoming air is provided round the gas outlet pipe. The air then enters the vaporizer which is formed by the annular space between the internal circular shell 7 and the external shell 8. A supply of heated water is continually fed from the funnel to the external surface of the internal shell 7 and is evaporated thereby. There is consequently an annulus 9, which is always kept full of steam and hot air while the plant is at work. As soon as the entering air, which has already been heated by its passage through the air jacket 5, reaches the vaporizer it becomes saturated with steam in passing round the vaporizer on its way to the furnace supply.

A hand-operated fan 12 is used for blowing in air when starting up before the engine is working. The water supply to the furnace is heated by a "field" tube which depends into the gas-outlet pipe.

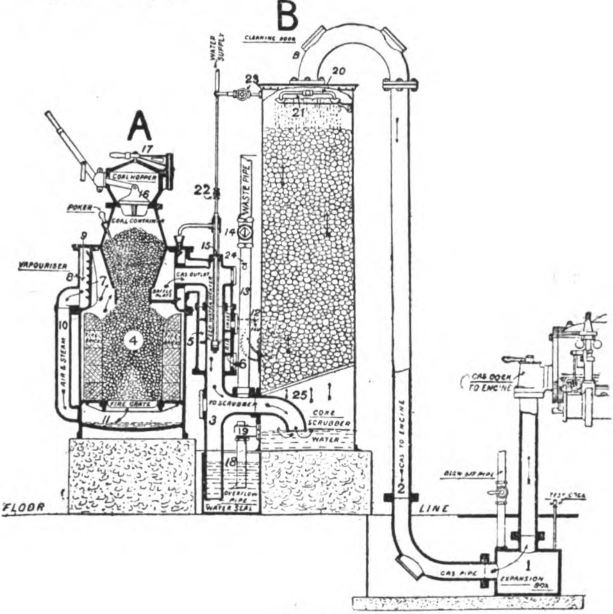


FIG. 43.

As it is essential that no air should enter the producer when at work excepting in the appointed way, double valves 16 and 17, which are opened one at a time, are provided for the coal hopper.

In connection with the coke-filled scrubber B, there is a seal pot 18 into which the overflow pipe 19 discharges the waste water from the scrubber. The water fed to the scrubber is spread over the whole surface of the coke by the distributing dish 21.

HUMPHREY EXPLOSION PUMPS.

A great many of the usual parts of an internal combustion engine and pump are dispensed with, and the system has been utilized in various ways for pumping large quantities of water or other liquid against a small head or pressure, or for dealing with smaller quantities in opposition to high pressures. The pump has also been employed to compress air at either high or low pressures.

Explosion takes place in a vertical chamber or cylinder, at the upper end of which are disposed inlet and exhaust valves. This combustion chamber is closed by the surface of a column of water instead of by the usual piston. The column of water oscillates, and the surface thus functions as a piston. The cycle of operations is either four or two stroke, and the usual cylinder cooling arrangements can be entirely dispensed with.

IGNITION.

Gas and petrol engines require means independent of the engine cycle for ignition. Electric ignition is now universally employed, the only exception being hot-tube ignition, which is still occasionally used in gas engines. The object is essentially to obtain a spark across an air gap in the cylinder by creating a sufficient momentary difference of voltage. This is effected by—

(1) The high-tension method in which a low-voltage current flowing in a primary circuit independent of the sparking circuit is suddenly broken and an induced secondary current of high voltage is sent across the gap in the sparking plug.

(2) The low-tension method, in which the electric circuit is suddenly broken in the combustion chamber by a small movement given to a lever.

Accumulator and Coil Ignition. The system is shown diagrammatically in Fig. 44. An induction coil shown in the upper part of the figure is associated with a contact breaker L, distributor M, and battery N to obtain a spark in each of the four cylinders of an engine in turn. The induction coil consists of (1) a core A, or bundle of soft iron wires, (2) a primary winding consisting of about three layers of insulated copper wire of about $\frac{1}{20}$ inch diameter, (3) a secondary winding of very fine wire containing 12,000 to 15,000 turns, (4) a condenser *b*, and (5) an automatic make-and-break device or trembler arranged in the primary circuit.

If the terminals B, C, are connected directly or indirectly to the battery N the primary or battery circuit is automatically made and broken by the trembler arm D and associated parts so rapidly as to give a continuous buzz. The condenser ensures that no spark shall occur when the primary circuit is closed, but a stronger spark occurs when it is opened, that is, a spark occurs at "break" but not at "make." The induction coil forms generally a compact separate unit or box with terminals.

In the construction of contact make-and-break shown at L a circular aluminium case includes a ring of insulating material in which is embedded flush with the surface a number of steel segments *d* spaced evenly round the circle, the number of segments being equal to the number of cylinders. Each segment is connected to the terminal C of the coil. The central axis *e*, usually the cam shaft, carries round at half-engine speed a

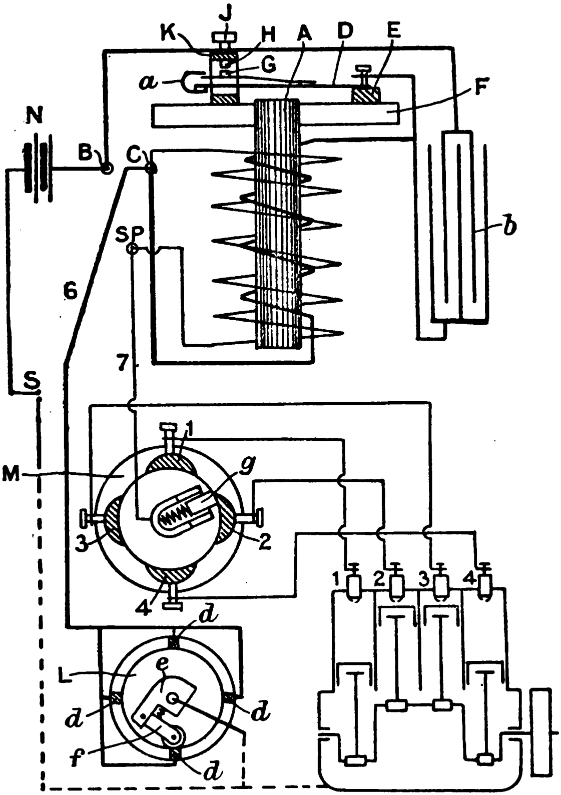


FIG. 44.—ACCUMULATOR AND COIL IGNITION.

metal arm *f* having a steel roller on its end which engages the steel segments in turn, closing the primary circuit on each occasion for a short but definite period. During each of these periods the trembler buzzes and gives a number of induced electrical impulses in the secondary circuit of sufficient voltage to give sparks in the cylinders. The distributor M is used to send these high-tension currents to the sparking plugs in turn. The distributor is similar in principle to the contact make-and-break.

The ignition system is put into or out of operation by a switch S. The centre of the contact make-and-break, the battery, and the external parts of the plugs are earthed, that is, connected to the engine or framework. The primary and secondary circuits will now be traced out for the case when a spark is required in the second cylinder.

Primary Circuit. Starting from the battery the current flows through terminal B, trembler D, contact E, primary coil, terminal C, conductor 6, steel segment *d* in contact breaker, roller on arm *f*, rotating axis *e* to "earth," and then returns through switch S to battery.

Secondary Circuit. Starting from the contact C, when the primary circuit is broken, the induced high-tension current flows through the secondary coil, the terminal SP, conductor 7, central rotating part of distributor M, carbon brush *g*, steel segment 2, and sparking plug 2 to "earth," and then returns through the contact-breaker L and conductor 6 to the terminal C.

Rotation of the low-tension contact make-and-break L and the high-tension distributor M at half-engine speed brings the sparking plugs successively into action in the order 1, 3, 4, 2, or 1, 2, 4, 3.

Magneto Armatures. These are of special shape applicable with suitable variations to both high and low-tension magnetos. A pictorial view of the iron core of a high-tension armature is shown in Fig. 45 and a sectional view of the core with the brass end pieces E, F is shown in Fig. 46. The core consists of a number of thin iron plates A insulated from one another and bolted together between two solid iron end plates B, C, the whole being so shaped as to leave a channel all the way round to receive the winding D.

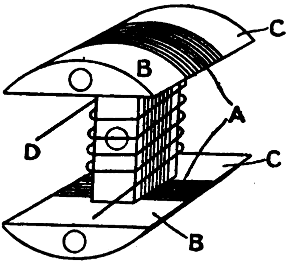


FIG. 45.—ARMATURE CORE.

When the core has been wound, circular brass end pieces E, F, adapted to run in ball bearings, are secured by screws.

The coil consists of insulated copper wire, each layer being also carefully insulated from those above and below it by layers of silk or enamel, so as to prevent leakage between wires or layers of wire. In a low-tension magneto about 150 turns of comparatively thick wire about $\frac{1}{20}$th inch diameter are employed, and

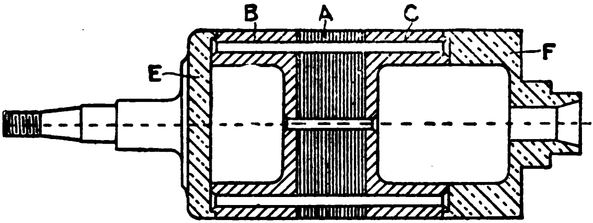


Fig. 46.—Magneto Armature.

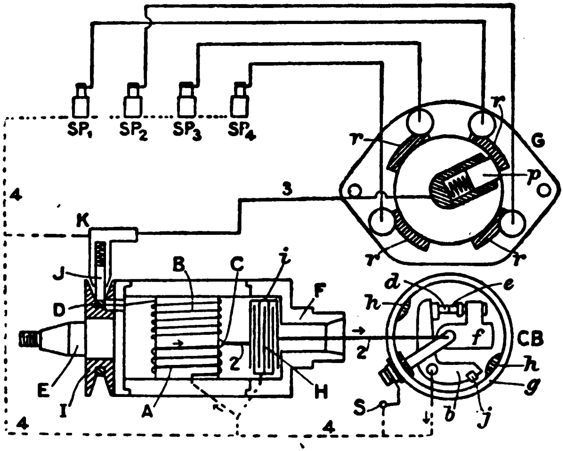


Fig. 47.—Magneto for Four-cylinder Petrol Engine.

A.	Primary Winding.
B.	Secondary Winding.
b.	Contract-breaker Lever.
CB.	Contract Breaker.
D.	Collector Ring.
d, *e*.	Platinum Points.
E, F.	Armature End Pieces.
f.	Insulated Block.
G.	Distributor.
g.	Contract-breaker Case.
H.	Condenser.
h.	Non-rotating Segments.
I.	Insulator for Collector Ring D.
J.	Collector Carbon Brush.
K.	Safety Gap.
p.	Distributor Carbon Brush
r.	Metal Segments.
S.	Switch.
SP_1, SP_2 SP_3, SP_4	Sparking Plugs.
2.	Primary Conductor.
3.	Secondary Conductor.
4.	Earth Connections.

this generates a current of 30 or 40 volts. The modern high-tension magneto has an inner winding of thick wire, and an outer winding of several thousand turns of much finer wire.

The armature in use is rotated between the poles of a permanent horse-shoe magnet and its windings cut the lines of force of the magnet and produce alternating currents in the winding, the maximum being obtained twice in each revolution when the laminated core A is nearly vertical. Two sparks can then be obtained per revolution of the armature, which therefore in a four-cylinder engine runs at crank-shaft speed, and in a six-cylinder engine at 1½ times crank-shaft speed.

High-tension Magneto Ignition. Fig. 47 shows diagrammatically the various parts of a magneto and its connection to the plugs of a four-cylinder engine, while Fig. 48 shows a section through a complete magneto.

The magneto consists of several main parts, as follows—

(1) The magnets Y and pole pieces.
(2) The rotor or armature.
(3) The contact-breaker CB.
(4) The distributor G.

One end of the primary winding A of the armature is earthed by connecting it direct to the iron core. The point C, where the primary and secondary windings join one another, is connected to the centre pin of the contact-breaker CB by a conductor 2. The other end of the secondary winding is led to a metal collector ring D carried to an insulating part I. A carbon brush J carries the high-tension current to the distributor G, from which it is conducted to the sparking plug in that one of the four cylinders in which the mixture is fully compressed. A compact condenser H consisting of a number of circular sheets of tinfoil, is arranged inside the brass end piece F of the armature. One set of condenser plates is connected to the low-tension conductor 2, while the other set of alternate plates is earthed by means of a connection to the end-piece F. The contact-breaker CB runs at the same speed as the armature and is arranged to break the primary circuit twice in each revolution of the armature. Its construction is as follows: A bent lever *b* is pivoted on a pin R carried by the disc on the end of the armature spindle, so that the lever is electrically connected to the armature core, and is thus earthed. The long arm of the bent lever carries a platinum point *d*, which is pressed by a spring into contact with a similar adjustable point *e* carried on a block *f*, insulated from the disc and connected to the conductor 2. The whole of the parts just described rotate with the armature inside a circular case *g*, on the inside of which are two projecting steel segments or rollers *h*. Twice in each revolution the projecting plug *j* on the short arm of the bent lever *b* knocks against a segment *h*, and the platinum points are separated sharply, thus breaking the primary circuit.

The high-tension lead 3 is connected in turn to sparking plugs SP_1, SP_2, SP_3, and SP_4 through the medium of the distributor G, running in the case of a four-cylinder engine at half crank-shaft speed.

The primary circuit consists of the primary winding A, the conductor 2, the block *f*, platinum points *de*, lever *b*, pivot of lever, and the earth return 4 to the primary winding A.

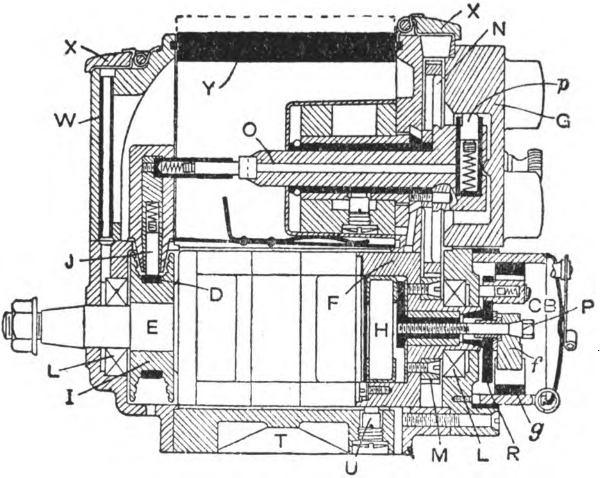


FIG. 48.—THOMSON-BENNET MAGNETO.

CB. Contact Breaker.
D. Collector Ring.
E, F. Armature End-pieces.
f. Insulated Block.
g. Contact-breaker Case.
G. Distributor.
H. Condenser.
I. Insulator for Collector Ring D.
J. Collector Carbon Brush.
L. Ball Bearings.
M, N. Gear Wheels.

O. Distributor Spindle.
P. Contact-breaker Screw Fastening.
p. Distributor Carbon Brush.
R. Disc on Armature Spindle.
T. Magneto Base.
U. Earthing Contact for Armature.
W. Aluminium End Plate.
X. Lubricating Covers.
Y. Magnet.

The secondary circuit consists of the secondary winding B, the collector ring D, carbon brush J, conductor 3, carbon *p* of distributor, one of the wires leading to a sparking plug, spark gap, earth return 4, and primary winding A.

The manner in which the magneto produces a spark will now be considered. Just before the points *d*, *e* of the contact-breaker are separated the rotation of the armature between the magnet poles causes a flow of current through the winding A complete primary circuit. At the same time there is ed in the secondary winding an electrical pressure which ufficient by itself to cause a spark, this effect being due to urning of the secondary winding in the magnetic field. n the contact-breaker opens the primary circuit the con- ser prevents sparking across the points *d*, *e*. The breaking he primary and the consequent stoppage of the current in primary winding induces a momentary current in the ndary circuit B. These two effects combined result in a mentary electrical pressure of over 10,000 volts in the second- winding, and this is sufficient to cause a spark across the of one of the sparking plugs. Earth—that is to say, the ine or frame—forms the return for both the primary and secondary circuits.

o sum up, the two principal effects which together result the production momentarily of a very high electrical pressure the secondary circuit are—

(1) The pressure in the secondary winding due to its movement ross the lines of force in the magnetic field.

(2) The induction effect in the secondary circuit due to the eck of the flow in the primary circuit.

In the event of the voltage in the secondary circuit becoming xcessive the current may escape to earth through the safety ap K; that is, the sparking plug is short-circuited. This voids the risk of an excessive pressure in the secondary circuit reaking down the insulation of the armature winding.

The magneto may be put out of operation by a switch S, hich will then connect the block *f* directly to earth, so that the rimary current cannot flow through the points *d*, *e*.

The two spindles on the ends of the armature run in ball earings L, the spindle or end-piece E being driven from the gine and the end-piece F driving the distributor spindle O half the speed of the armature through gears M, N, and carry- the rotating parts of the contact-breaker. The pin P ves to connect the block *f* electrically with the armature ding, to clamp the block to the disc E, and to hold the disc he end of the armature spindle. The shell carrying the steel nents of the distributor is secured to the aluminium cover- by means of two set screws. An aluminium cover plate so closes the driving end of the magneto. ensure constant electrical connection between the armature he base T for earthing purposes a carbon contact U is ed.

spark may be advanced or retarded by adjusting the ngularly by means of the arm V.

tension Oscillating Magnetos are often used on gas The magneto is very similar to the high-tension magneto, but the armature, instead of being rotated sly, is oscillated by a trip gear in which a tripping ted to the armature spindle is actuated by a cam on ed shaft, and when the cam releases the arm, springs armature and tripping lever to the central position.

No distributor is required, the sparking plug being connected directly to the secondary terminal.

Low-tension Magneto Ignition. Oscillating low-tension magnetos are used on many gas engines. The magneto is similar to the high-tension oscillating magneto, but the armature has only one winding, one end of which is earthed through the brass base while the other end is connected to the contact-breaker in the ignition block. No contact-breaker in the magneto is necessary.

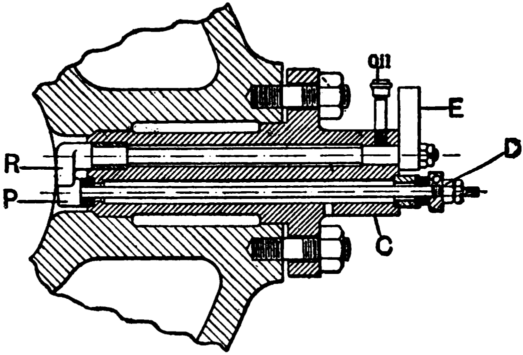


FIG. 49.—LOW-TENSION IGNITION BLOCK.

A Crossley low-tension ignition system, as secured to the outside of the combustion chamber of a gas engine, consists of—

(1) The ignition block (Fig. 49).

(2) The magneto and its operating mechanism.

The current from the magneto is conducted to the outer end of the contact pin D (which is supported in the ignition block C by insulating pieces composed of mica washers) to the sparking point P, the contact-breaker R (consisting of an arm on the end of a rocking spindle, the outer end of which carries the ignition lever E), the ignition block C, and the engine. When the current is flowing strongly the ignition lever E is suddenly turned so as to introduce a gap in the circuit between R and P. The arm E is operated at about the end of the return movement of the magneto after it has been tripped. In oscillating magnetos the quality of spark is independent of engine speed.

THE DIESEL ENGINE.

The Diesel Motor, which is now built exclusively as an oil engine, operates to all intents and purposes on the constant pressure cycle, and owes its excellent efficiency mainly to the employment of a high compression ratio and to the means employed for combustion of the fuel by which the lowest and cheapest grades of oil can be satisfactorily employed. A theoretical diagram for the Diesel cycle is shown in Fig. 1

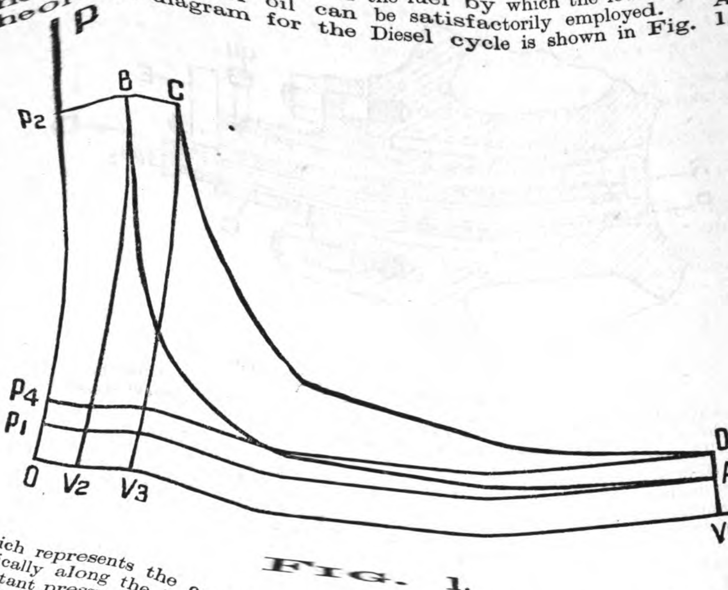


FIG. 1.

which represents the compression of air being effected
batically along the line AB, after which heat is taken
constant pressure P_2 along the line BC, following which e
sion occurs adiabatically to the point D at the pressur
when there is rejection of heat to the exhaust at con
volume, to the point A, where the pressure is P_1. The
pression ratio which is represented by the ratio betwe
and V_2 is usually about eleven to one, instead of about
one, which is common with gas engines. In actual pr
the indicator cards obtained from these Diesel engines

somewhat from that shown in Fig. 1 as combustion is not prolonged to the point C, and does not take place wholly at constant pressure.

The method of operation in a Diesel engine is as follows, referring to the vertical type, since the large majority of motors operating on this cycle are of the vertical design. During the first downward stroke of the piston air at atmospheric pressure is drawn into the cylinders through a mechanically operated valve, and this action is followed by the compression of the pure air thus drawn in on the upward stroke, all valves being closed during this movement. The compression is carried to a pressure between 450 and 500 lbs. per square inch, although in a few of the more recent designs, this pressure has been diminished to some extent.

Just before the piston reaches the top of its compression stroke, the fuel inlet valve is opened mechanically and the fuel is injected into the combustion chamber, usually by a blast of compressed air at a pressure of about 800 lbs. per square inch, the spray valves being of a special type, causing the fuel to be thoroughly atomized during its passage through the nozzle. Owing to the high temperature of the compressed air in the combustion chamber (about 1,000 degrees Fahr.), and the efficient pulverization of the fuel, combustion immediately takes place on the entry of the oil, and the fuel valve is closed usually after the piston has travelled over about one-tenth of its full stroke. Combustion continues for a very short period after the fuel valve is closed, and the pressure generated gives the piston its downward working stroke. Shortly before it reaches the bottom dead centre, the exhaust valve is opened and the pressure drops rapidly, all the products of combustion being expelled through the exhaust valve during the next upward stroke of the piston, after which the cycle recommences.

In Diesel motors of the two-stroke type, the same series of operation is carried out in two strokes of the piston, instead of four. At the end of the working stroke, the cylinder is full of air at approximately atmospheric pressure, and this is compressed on the upward stroke to from 450 to 500 lbs. per square inch. As before, the fuel is injected into the combustion space, just before the piston reaches the top of its stroke; combustion and expansion follow, on the downward stroke, and when the piston has passed through about three-quarters of its travel, it commences to uncover the exhaust ports which are cut in the cylinder, and exhaust commences. Shortly after this takes place scavenging air at a pressure of about 4 lbs. per square inch, supplied by scavenging pumps driven off the engine, enters the cylinder either through mechanically operated valves in the cylinder head, or through scavenging ports in the bottom of the cylinder on the opposite side to the exhaust ports. This scavenging air thus expels all the products of combustion and fills the cylinder with pure air ready for compression on the next upward stroke of the piston.

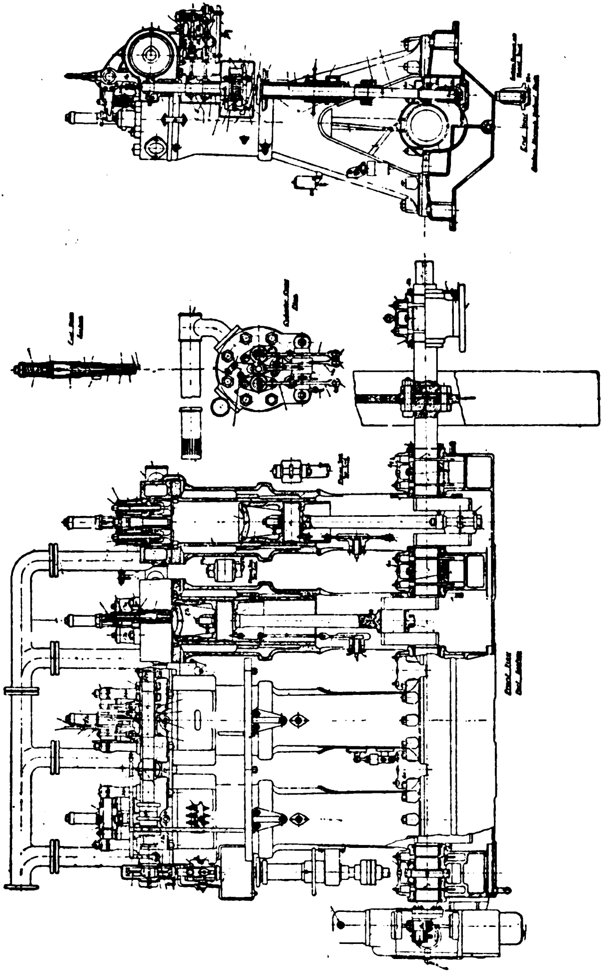


FIG. 2.—WILLANS DIESEL ENGINE.

For the supply of compressed air for injecting the fuel, and also for purposes of starting up the engine, an air compressor is fitted to Diesel engines, usually driven direct off an extension of the crankshaft, whilst in motors of the two-stroke type the scavenging pump has also to be provided. For starting purposes, air reservoirs are supplied, these being charged from the air compressor. Some Diesel engines are, however, now built (especially for marine work) in which solid injection is employed, the oil being pumped by means of a high-pressure mechanical pump at a pressure of about 4,000 lbs. to 7,000 lbs. per square inch into the cylinder. For stationary work, however, air injection is almost universally adopted.

Four-cycle engines are somewhat more efficient than two-cycle motors, and have a fuel consumption varying from about 0·38 lbs. to 0·44 lbs. per b.h.p. according to the size, whilst the consumption of two-cycle engines is from 0·42 to about 0·47 lbs. per b.h.p. The mechanical efficiency of a four-cycle motor is generally about 74 to 80 per cent., whilst that of a two-cycle engine varies from 68 to 75 per cent. In the latter class of motor, port scavenging is now generally preferred to the employment of valves, owing to the simplification of design, and the fact that numerous valves in the cylinder heat, usually lead to cracks developing in the covers.

TYPES OF DIESEL ENGINE.

For stationary purposes the Diesel engine is built both of the two-cycle and four-cycle type, although the former is seldom constructed under powers of about 700 h.p. The largest Diesel motor ever constructed for commercial purposes is one of 3,600 h.p. in six cylinders, operating on the two-cycle principle, built by Sulzer Brothers, of Winterthur.

Both high-speed and low-speed engines are manufactured, the latter usually running at about 260 to 350 r.p.m., and the former from 150 to 200 r.p.m., according to the output. The fuel consumption of the high-speed motor is slightly greater than that running at a lower speed of revolution, but the cost of construction is somewhat reduced, and they are more suitable in certain cases for driving dynamos.

In Fig. 2 is shown a typical Willans Diesel engine of the four-cylinder type, manufactured by Willans and Robinson, Ltd., Rugby. This is a slow-running set, and the design represents what may be termed, normal Diesel engine practice. The cylinder cover liner and jacket are separate castings, and all the valves are arranged in the cylinder head, a system that is adopted almost universally in Diesel engines, except one well-known class of motor built in America, in which the exhaust and inlet valves are horizontal. The horizontal cam-shaft, provided with cams for actuating the starting, inlet, exhaust, and fuel valves, is driven from the vertical governor shaft seen on the left. This latter is itself driven from the crankshaft through gearing. The vertical shaft also operates the fuel pump and drives the governor, which controls the speed of

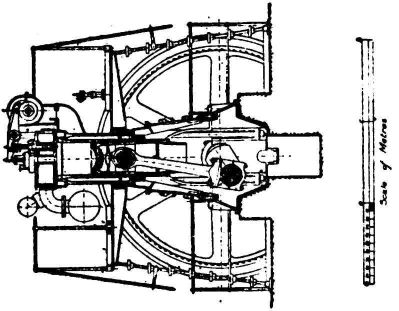
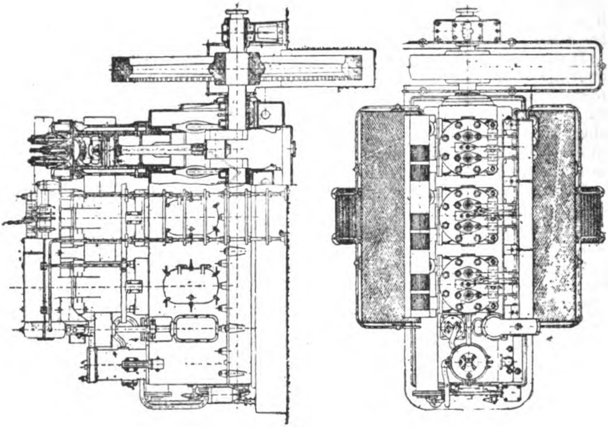


FIG. 3.—BURMEISTER AND WAIN HIGH-SPEED DIESEL ENGINE.



the engine by varying the opening of the suction valves of the fuel pumps during the pressure stroke, so that part of the fuel is by-passed, should the speed rise too high. The compressed air for injection purposes is provided by the air compressor on the end of the engine, this being of the Reavell quadruplex type. Like all other Diesel engines, constructed in powers under 500 h.p., the Willans motor is of the trunk piston type.

Fig. 3 shows a high-speed Diesel engine constructed by Burmeister & Wain. One of the main points of difference is that the crank chamber is wholly enclosed, forced lubrication being employed throughout, and there are large doors through which access to the connecting rod and main bearings is obtained. The camshaft is also enclosed with a casing, but otherwise in general design there are few modifications of importance, the separate cylinder head and liner being retained. The camshaft is arranged in the same position as in the previous engine, and is driven in a similar manner through a vertical governor shaft, on which the governor is fitted, but instead of the Reavell compressor one of the vertical three-stage type is employed. It will be noticed that the cylinders are supported, not only by the crankcase, but also through long vertical steel bolts, passed right from the bottom of the bedplate, to the cylinders.

In Fig. 4 is shown a Diesel engine of somewhat different construction, built by the Werkspoor Co., of Amsterdam. This is designed mainly for marine work, but is also built along the same lines for stationary purposes. It is of the crosshead type, and the piston is water-cooled, a system that is seldom employed in motors under about 800 or 1,000 h.p. This is effected by means of pump circulation and a telescopic pipe attached to the piston dipping into the long tube seen in the illustration. The crank chamber is enclosed, and is provided with light plates, which can be readily removed for inspection, and the cylinders are supported by long through bolts, extending from the bedplate to the top of the cylinder covers. For strengthening purposes, cross bracing is also usually adopted with this type of motor.

The three engines just described are of the four-cycle type, and in Fig. 5 is shown a two-cycle motor built by Messrs. Sulzer Bros. In this the system of scavenging by ports in the cylinder walls is adopted, so that there are only two valves in the cylinder cover, one for the admission of fuel, and one for starting purposes. The exhaust ports leading to the exhaust pipes will be seen on the right, whilst on the left will be noticed two sets of ports, one arranged above the other. The bottom row is always open to the scavenging trunk, to which air at a pressure of about 4 lbs. per square inch is supplied by the scavenging pumps driven off the engine. Immediately the piston, on its downward stroke, commences to uncover the bottom row of ports, the scavenging air enters the cylinder, from the scavenging trunk and tends to clear out the products of combustion through the exhaust ports. The admission of

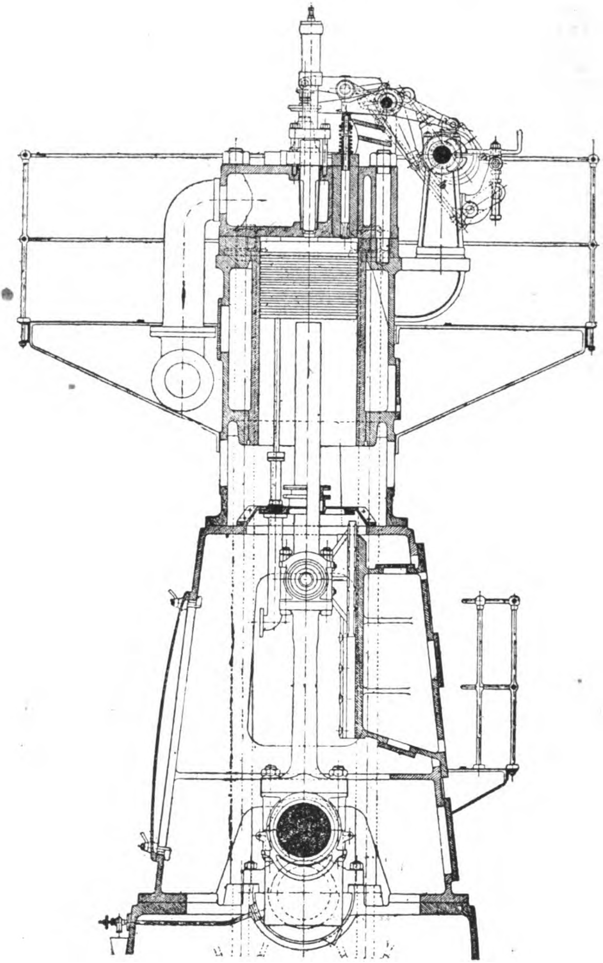


FIG. 4.—WERKSPOOR DIESEL ENGINE.

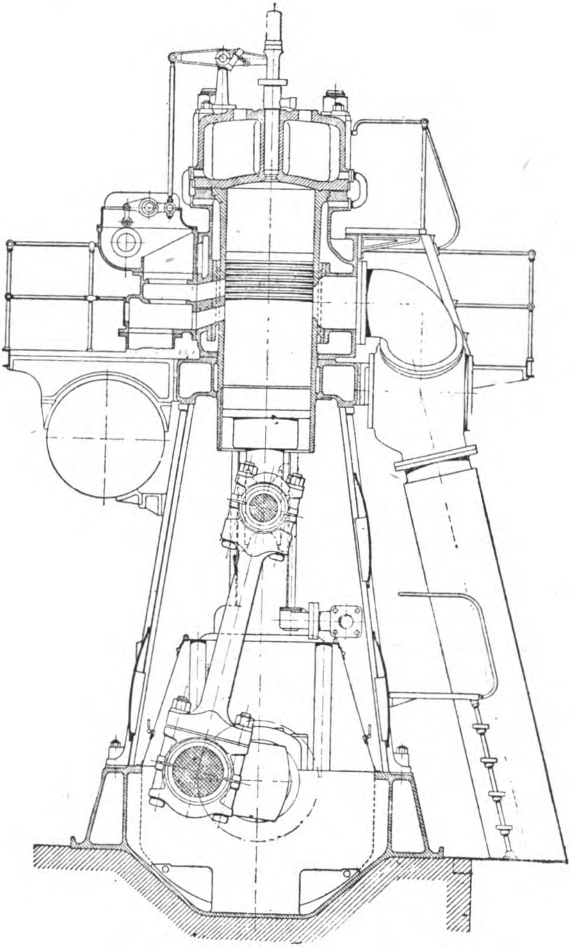


FIG. 5.—SULZER TWO-CYCLE ENGINE.

scavenging air from the trunk to the top row of ports, however, is controlled by a rotary valve actuated from a horizontal shaft and the air cannot enter the cylinders through the ports until the valve is lifted at the period when it is required to open. This valve is actuated when the piston on its upward stroke is just closing the bottom row of scavenging ports, and the extra scavenging air which thus enters helps not only to expel the exhaust gases through the exhaust ports but also continues to admit scavenging air into the cylinder after the exhaust ports have been closed by the piston in the course of its upward stroke. The result of this is to ensure that the cylinder becomes full of pure air, and it is found that improved combustion is effected as compared with an engine with a single row of scavenging ports. Engines of this design are built in powers from 420 h.p. up to 3,600 b.h.p., but in the larger sizes the trunk piston is replaced by the crosshead arrangement.

Marine Diesel Engines. Various types of Diesel engines are built for marine work, but they are, generally speaking, based on the same main principles as those which have been described. They are, however, invariably made directly reversible, and this is usually accomplished by having two cams for the operation of each valve, mounted side by side on the camshaft. One of these cams is shaped for actuating the valve lever for running in the ahead direction, and the other for the astern operation. When it is desired to go astern the valve levers are lifted off the cams—either by means of a hand-wheel or a Servo motor—the camshaft is moved longitudinally, until the astern cam comes below the valve rollers, when the latter are then dropped on to the cams and the engine started up by compressed air. Various modifications of this arrangement are adopted by different manufacturers, to meet the special requirements of their engines.

Amongst the marine Diesel engines which are being built in Great Britain are one or two of unusual design. That developed by Doxfords has opposed pistons, the air being compressed between the two pistons, on their inward stroke, and the space between them at the dead centre forms the combustion chamber into which the fuel is injected in the usual way. The scavenging air is admitted to ports at the top of the long cylinder, and the exhaust takes place, at the bottom through other ports, both sets being uncovered by the respective pistons on their outward stroke. In this engine as in the Vickers' four-cycle type, solid injection of fuel is utilized, instead of the employment of highly compressed air as is usual with most Diesel engines. The Doxford motor operates on the two-cycle principle.

DESIGN OF DIESEL ENGINES.

The cylinder liners, jackets and covers, are almost invariably made of cast iron, except in submarine motors, and the attempts which have been made to use cast steel covers, in one or two large marine engines have not proved successful, and have been abandoned. The thickness of the cylinder liner usually varies between the following limits—

$T = 0{\cdot}09$ to $0{\cdot}10$ D for four-cycle engines, and
$T = 0{\cdot}1$ to $0{\cdot}13$ D for two-stroke engines,

where

T is the thickness of the liner and
D the diameter of the cylinder.

The indicated horse power of a Diesel engine is

$$\text{IHP} = \frac{P \times \frac{L}{12} \times A \times M \times N}{33{,}000}$$

where P = mean effective pressure in lbs. per square inch;
L = length of stroke in inches;
A = piston area in square inches;
M = number of working strokes per minute;
N = number of cylinders.

In a four-cycle engine $M = \frac{n}{2}$

where n = RPM and in a two-cycle engine M = n.

The mean effective pressure in a Diesel engine usually varies between 90 and 100 lbs. per square inch in a four-cycle motor, and 80 to 95 lbs. per square inch in a two-cycle motor of the slow-speed type, being somewhat lower in high-speed sets.

The mechanical efficiencies of Diesel engines are approximately as follows—

Type.	*Efficiency per cent.*
Four-cycle slow speed	74 to 80
Two-cycle slow speed	68 to 75

High-speed engines are about 4 or 5 per cent. lower.

A good working formula for the brake horse power and of average *four-cycle Diesel* engine is—

$$\text{BHP} = {\cdot}00008\,D^2LnN,$$

where D = cylinder diameter in inches;
L = length of stroke in inches;
n = revs. per minute;
N = number of cylinders.

In two-cycle engines—

$$\text{BHP} = 0{\cdot}00014\,D^2\,LnN.$$

The ratio of bore and stroke varies from about 1·4 to 1·6 in four-cycle engines, and 1·4 to 2 in two-cycle engines, of the slow-speed type. For high-speed sets the ratio varies from unity to 1·2.

The piston speeds usually adopted are given below—

Type.	*Piston Speed Ft. per minute.*
Four-cycle slow speed	650–800
„ high „	800–1000
Two-cycle slow speed	700–800
„ high „	850–1000

Lloyd's Rule for Diesel engine crank shafts is

$$d = \sqrt[3]{3D\ A}^2$$

Where d = diameter of crankshaft in centimetres;
D = „ cylinder „
A = constant in following table;
H = stroke of piston in centimetres;
L = distance between two adjacent bearings in centimetres.

No. of Cylinders.	*A*
1, 2, and 3 ..	$0{\cdot}09\,H + 0{\cdot}035\,L$
4	$0{\cdot}10\,H + 0{\cdot}035\,L$
5	$0{\cdot}11\,H + 0{\cdot}035\,L$
6	$0{\cdot}13\,H + 0{\cdot}035\,L$

STEAM AND STEAM ENGINES

THE UNIT OF HEAT.

For most practical purposes the British Thermal Unit (B.Th.U.) may be taken to be the heat required to raise the temperature of 1 lb. of water 1° F. For precise work allowance must be made for the fact that the specific heat of water varies with temperature. Rankine's definition of the B.Th.U. is based upon the heat required to raise the temperature of 1 lb. of water from 39° to 40° F.; Peabody's definition is based upon the temperature range 62° to 63° F.; and Marks and Davis take as the unit $\frac{1}{180}$ of the heat required to raise 1 lb. of water from 32° to 212° F.

The British Thermal Unit or pound-degree-Fahrenheit unit is the one generally used by practical engineers, but the pound-degree-Centigrade Heat Unit or Centigrade Heat Unit is also used by some engineers; this is the amount of heat required to raise the temperature of 1 lb. of water by 1° C., and is generally denoted by the abbreviation C.H.U.

1 C.H.U. = $\frac{9}{5}$ B.Th.U. = 1400 ft. lbs.
1 B.Th.U. = $\frac{5}{9}$ C.H.U. = 778 ft. lbs.

Other useful relationships are—

1 horse power = 33,000 ft. lbs. per min.
= 0·746 kilowatts.
1 kilowatt = 1·3404 h.p.
1 horse power-hour = 1,980,000 ft. lbs.
= 2,545 B.Th.U.
1 kilowatt-hour = 2,655,000 ft. lbs.
= 3,410 B.Th.U.

STEAM.

Dry, Saturated Steam. Steam is said to be dry and saturated when the water from which it is formed is just completely evaporated but without being raised above the boiling point corresponding to the pressure concerned. The *boiling point of water* and the temperature of steam in contact therewith increases with the pressure to which the water and steam are subjected (see Steam Tables). At normal atmospheric pressure (14·7 lbs. per sq. in.) the boiling point is 212° F.

The *specific heat* of dry, saturated steam is 0·48. *Superheated Steam* is steam at a temperature above the boiling point of water at the pressure concerned. The density of superheated steam varies with temperature; when considerably superheated, the density of steam is approximately 0·62 × that of air at the same temperature and pressure.

Wet Steam. Dryness Factor. Wet steam consists of steam carrying particles of water which are at the boiling point, but are not evaporated. The dryness factor of wet steam = q = Weight of saturated steam/Total weight of steam and water mixture. The total heat of wet steam = sensible heat* + (q × latent heat) = ($h + q$L) B.Th.U. per lb. If the

* i.e. heat absorbed from 32° F. to boiling point.

Pressure of wet steam be reduced by throttling down, without doing work, the dryness of the steam is increased; but if the pressure of the steam be reduced by expansion, work being done, the dryness factor is usually reduced. Therein lies the bjection to too great expansion in a single cylinder; the walls f the cylinder become both cold and wet, and serious con-ensation occurs on the next admission (see also Compounding nd Uniflow Engines).

Steam Tables. The steam tables on pp. 580 give those phy-cal characteristics of *saturated* steam which are required in lculations, etc., relating to steam engines. Provided that e steam is saturated, any one of these physical characteristics ffices to determine its condition, the other characteristics ing then as shown in the steam table. The student will find instructive to plot the various characteristics as functions the temperature.

The total or *absolute pressure* of the steam is a definite cha-cteristic but the *gauge pressure* depends upon the atmospheric essure.

bsolute pressure = Gauge pressure + Atmospheric pressure.

For most purposes and in most places it is sufficiently accurate assume the atmospheric pressure = 14·7 lbs. per sq. in., t at altitudes and for the purpose of accurate trials the actual mospheric pressure must be determined from the reading of e barometer.

If steam tables are not available the absolute pressure p lbs. r sq. in. corresponding to any temperature t° F. (or the inverse eration) may be determined approximately from—

$$\log p = 5\left(\frac{t - 212}{t + 367}\right) + \log 14{\cdot}7$$

The *total heat* of the steam is the sum of the heat required to aise water from 32° F. to the boiling point at the temperature con-cerned and the latent heat required to convert the boiling er to dry, saturated steam. The formula given by Marks Davis for the total heat of saturated steam at t° F. is—

$$H = 1150{\cdot}3 + 0{\cdot}3745\,(t - 212) - 0{\cdot}00055\,(t - 212)^2$$

re H = B.Th.U. above 32° F. and t = steam temperature in

A simpler but less accurate formula is—

$$H = 1082 + 0{\cdot}305\,t$$ which corresponds to the formula

tent heat—

$$L = 1114 - 0{\cdot}7\,t.$$

ume per Lb. of Saturated Steam. The relation between solute pressure, p lb. per sq. in., and the volume, in cu. ft., of dry saturated steam is given approximately by—

$$p\,V^{\frac{17}{16}} = 479$$

mple. If $p = 160$, $V^{\frac{17}{16}} = 2{\cdot}994$ and $\frac{17}{16}\log V = \log$ ence $\log V = \frac{16}{17} \times 0{\cdot}4762 = 0{\cdot}4483$ and $V = 2{\cdot}807$ mpared with 2·79 cu. ft. as given by steam tables.)

g the temperature and pressure of steam at and about e for which the volume V cu. ft. per lb. is required, V may be found as follows. The volume of 1 lb. water is 0·016 cu. ft., hence the change of volume on evaporation is (V − 0·016) cu. ft., and the work done by expansion through a pressure difference dP is dP (V − 0·016) ft. lb. = dP. (V − 0·016)/778 B.Th.U. The work done during a temperature drop dT is Q.dT/T where Q = heat units imparted to the working fluid at temperature T. In this case Q = latent heat L, hence dP (V − 0·016)/778 = L.dT/T whence—

$$V = \frac{778\,L}{T} \cdot \frac{dT}{dP} + 0{\cdot}016$$

Example. Find the specific volume of dry saturated steam at 100 lbs. per sq. in. From steam tables L = 888, T = 787·8, and the temperature difference for 5 lbs. per sq. in. (720 lbs. per sq. ft.) pressure difference = 3·7° F., hence $dT/dP = 3{\cdot}7/720$ $= 0{\cdot}00514$ approx. Therefore $V = \frac{778 \times 888}{787{\cdot}8} \times 0{\cdot}00514$ $+ 0{\cdot}016 = 4{\cdot}5$ cu. ft. per lb. (approx.).

The *entropy of water* φ_w from 32° F. to boiling point = $\log_e$ T/492 very nearly, and = $h/\frac{1}{2}(492 + T)$ approximately; where T = boiling point (° F. absolute) at the pressure concerned, and h = water heat or sensible heat = (T − 492) very nearly, hence $\varphi_w = 2\,(T - 492)/(T + 492)$ roughly.

The *entropy of evaporation* φ_s of dry saturated steam = Latent heat/Absolute temperature of steam.

The *external work done* in the conversion of 1 lb. water to 1 lb. of dry saturated steam is frequently given in steam tables and may be calculated from: Work done = $\sigma = 144\,p\,(V - v)/778$ B.Th.U. per lb.; where p = absolute pressure in lb. per sq. in.; V = volume per lb. of steam at pressure p; v = volume of 1 lb. of water at 32° F. = 0·016 cu. ft. approx.

The external work done increases with the temperature and pressure of the steam, but the latent heat decreases, hence the ratio σ/L increases. The theoretical maximum thermal efficiency of the old atmospheric engine, working without expansion, was σ/L; the actual efficiency was much lower owing to cylinder cooling, etc.

The *specific heat of steam* at constant pressure is generally taken to be 0·48, so that the heat required to superheat dry saturated steam from the temperature t of its formation to a temperature t' is: $0{\cdot}48\,(t' - t)$ B.Th.U. per lb.

INTERNAL ENERGY OF STEAM.

The internal energy of 1 lb. of dry saturated steam is the total heat (H) minus the heat equivalent (σ) of the external work done in raising 1 lb. of water from 32° F. and converting it to d.s. steam. In the case of wet steam we must allow for the dryness factor q; and in the case of superheated steam we must allow for the heat required to raise the temperature of the steam and for the external work done in increasing the volume of the steam during superheating; the heat equivalent of this work is $\frac{1}{778}\int p.dV$.

We have thus the following expressions—

	Total heat of formation above 32° F. B.Th.U. per lb.	Internal Energy Steam above 32° F. B.Th.U. per lb.
(*a*) Dry saturated steam	$(h + \mathrm{L})$	$(h + \mathrm{L}) - \sigma$
(*b*) Wet steam	$(h + q\mathrm{L})$	$(h + q\mathrm{L}) - q\,\sigma$
(*c*) Superheated steam	$(h + \mathrm{L}) + 0{\cdot}48\,(t' - t)$	$(h + \mathrm{L}) + 0{\cdot}48\,(t'-t)-(\sigma + \frac{1}{778}\int p.d\mathrm{V})$

Under all circumstances—

Heat supplied = Change in internal energy + External work done.

EXPANSION OF STEAM.

If a volume v_1 of dry saturated steam at pressure p_1 expands to a lower pressure p_2, the volume will be less than that calculated from the volume per lb., as given by steam tables, in the ratio of the dryness factor q. The actual volume $v_2 = q \times$ "tabular" v_2. The internal energy of the steam, I_1 at $p_1\,v_1$ and I_2 at $p_2\,v_2$, can be calculated from the above expressions. The work W done externally by the expansion is $\int p.d\mathrm{V}$, and can be calculated if the law of the expansion is known. Otherwise the shaded area Fig. 1 may be planimetered. Then :

Heat supplied to the steam during expansion $= (\mathrm{I}_2 - \mathrm{I}_1) + \mathrm{W}$.

If the sensible heat h and total heat H per lb. be plotted against absolute pressure we obtain two lines (Fig. 2), the intercept between which represents the latent heat to the

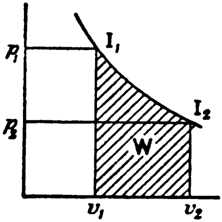


FIG. 1.—CHANGE IN INTERNAL ENERGY OF AND WORK DONE BY STEAM DURING EXPANSION.

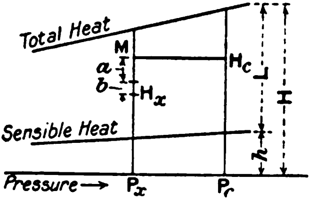


FIG. 2.—CHANGE IN TOTAL HEAT DURING EXPANSION.

scale chosen. Suppose that the pressure at cut-off is P_c and that the dryness fraction is q_c, then : Total heat at cut-off $= H_c = h_c + q_c L_c$. If the pressure of the steam be reduced to P_x without doing external work the total heat remains the same (point M, Fig. 2), i.e. $h_x + q_x L_x = H_c$, but both h and L are now smaller than before hence the dryness fraction must be higher, i.e. the dryness fraction is increased by throttling down the pressure without doing external work.

On the other hand, if external work be done equivalent to heat a and if the heat loss by radiation be b we reach the point H_x, Fig. 2, and $h_x + q_x L_x = H_x$, which corresponds to a lower dryness fraction. If heat be added to the cylinder charge from a jacket, H_x will be higher than is otherwise the case, and the dryness fraction may actually be higher than at cut-off.

ENTROPY.

When a body receives an amount of heat H at an absolute temperature T, its entropy is increased by the amount H/T, usually denoted by φ. In general : Entropy $= \int \frac{dH}{T}$ and $d\varphi = \frac{dH.}{T}$ Constant entropy involves constant H and is thus associated with adiabatic* change of state. In cases where an interchange of heat occurs, the heat change dH is given by $dH = T.d\varphi$, i.e. by the product of the absolute temperature into the change of entropy. In other words, the area under a curve, plotted with absolute temperature as ordinates and entropy as abscissae, represents the heat absorbed or emitted in passing from one point to another on the curve.

The *entropy of water* $= \log_e (T_2/T_1)$, assuming the specific heat of water at constant pressure to be 1·0 at all temperatures. In the case of φ_{water} from 32° F. to the boiling point the formula becomes $\varphi_{water} = \log_e (T/492)$ or approximately $h/\frac{1}{2}(492 + T)$.

The *entropy of vaporisation* of dry saturated steam (given in the more complete steam tables) = Latent heat/Absolute temperature of steam.

The *entropy of superheated steam* at constant pressure $= 0{\cdot}48 \log_e T_1/T_2$.

ENTROPY CHART FOR STEAM.

Referring to Fig. 3, entropy for water and steam is reckoned from a temperature 32° F. The entropy of water changing its temperature under constant pressure is $\varphi_w = k_p \log_e (T_2/T_1)$. Taking k_p, the specific heat of water at constant pressure, to be 1·0 (which is not quite true at all temperatures), and taking $T_1 = 492°$ F. absolute, we have $\varphi_w = \log_e (T/492)$ approximately. From this equation the *water line* OA is plotted, which shows the increase in entropy corresponding to a temperature change, e.g. 0·17 from *a* to *b*. The heat addition corresponding to this temperature change is given by the area enclosed by the curve *ab* and the verticals through *a* and *b* carried down to the horizontal through the *absolute zero* of temperature. The absolute zero line is not shown in Fig. 3, but for most practical purposes it is sufficiently accurate to take the heat increment from *a* to *b* = (Mean of the absolute temperatures corresponding to *a* and *b* × Increase in entropy from *a* to *b*) $= \frac{1}{2}[(120 + 460) + (230 + 460)] \times 0{\cdot}17 = 635 \times 0{\cdot}17 = 108$ B.Th.U.

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* In *adiabatic* expansion or compression, no heat enters the working fluid from the cylinder or from outside the cylinder, neither does any heat escape from the latter. *Isothermal* expansion or compression is carried out at constant temperature.

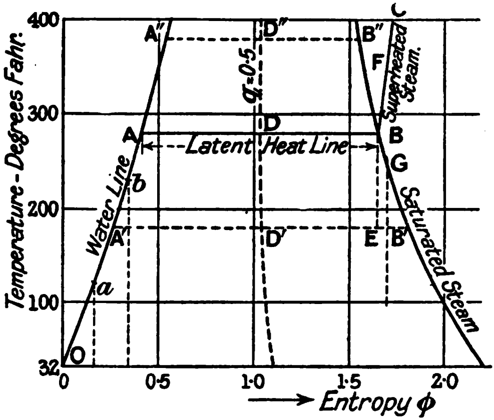

FIG. 3.—EXPLAINING ENTROPY DIAGRAM.

If the water be evaporated from any "state-point" A, the latent heat is added at the constant temperature corresponding to that point, and the entropy increases along the horizontal line AB. The area under this line (down to the line of absolute zero temperature) represents the latent heat. The right-hand ends of latent heat lines starting from different water temperatures mark out the entropy curve for saturated steam. The *entropy of evaporation* at any absolute temperature T is L/T where L is the latent heat of evaporation at that temperature.

If, starting from A, only half the latent heat is added, we obtain steam of dryness fraction $q = 0{\cdot}5$, represented by the point D. Similarly, points D′ D″ correspond to $q = 0{\cdot}5$ on the latent heat lines A′ B′, A″ B″ respectively and the curve through D″ DD′ is the entropy curve for steam of dryness fraction 0·5. Entropy curves for steam of other dryness fractions are obtained in the same way (see Fig. 4).

The entropy of steam superheated at constant pressure is given by $\varphi_s = 0{\cdot}48 \log_e (T_2/T_1)$, and is represented by a family of curves of which BC (Fig. 3) is typical (see also Fig. 4).

A convenient form of entropy chart is reproduced in Fig. 4,

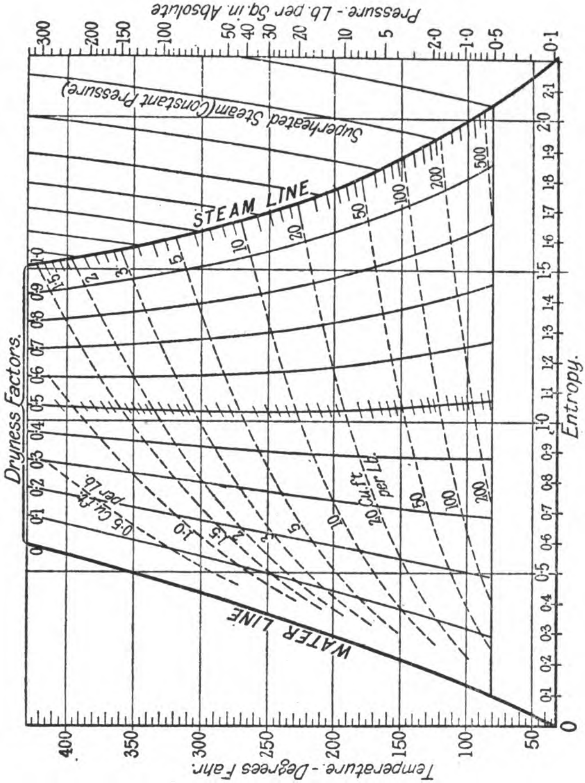

FIG. 4.—TEMPERATURE-ENTROPY CHART FOR STEAM.

this including a scale of absolute pressures and curves for the volume of steam per lb. For practical purposes the chart should be used on a much larger scale ; see, for instance, Marks and Davis's *Steam Tables*, Moyer's *Steam Turbine*, etc. A very useful entropy chart (12″ × 10″) is sold by H.M. Stationery Office.

USES OF THE ENTROPY CHART.

The fundamental relation between temperature, entropy, and quantity of heat makes it possible to determine from the entropy chart the third of these quantities when the other two are known.

Adiabatic changes are represented by vertical lines (constant entropy) on the chart. For example, saturated steam expanding adiabatically from a temperature of 280° F. (state-point B, Fig. 3) reaches state-point E at a temperature of 180° F. and is then of dryness fraction $= A'E/A'B' = 0{\cdot}89$. Similarly superheated steam at F, when expanded adiabatically, becomes saturated steam at G (245° F.) and wet steam of dryness fraction 0·92 at 180° F., and of 0·84 at 100° F.

The dryness fraction q_2 of steam after adiabatic expansion from an initial dryness fraction q_1 can be read directly from an entropy chart; if no chart is available q_2 may be calculated from—

$$q_2 = \frac{q_1\,(L_1/T_1) + \log_e\,(T_1/T_2)}{L_2/T_2}$$

If steam is to remain saturated whilst expanding from B to B′, heat must be added (this being represented by the area under the curve BB′ down to the absolute zero temperature line), and the entropy of the steam increases from 1·65 to 1·8.

Horizontal lines between the water and steam lines represent evaporation or condensation (as the case may be) at constant temperature (i.e. isothermal changes), the heat change being represented by the area under the line, measured down to the axis of absolute zero of temperature.

The dryness fraction and volume per lb. of steam corresponding to any state-point on the entropy chart can be read off from the constant-dryness and constant-volume curves (see Fig. 4), interpolating if necessary.

An entropy chart offers the best means of studying the loss of efficiency due to wiredrawing and pressure drop at release (see pp. 451, 452).

Entropy charts are indispensable for studying the heat changes in steam turbines; this matter is treated fully on pages 550–552.

MOLLIER DIAGRAM.

In this entropy diagram* the total heat, H, is plotted against entropy, φ. This curve is plotted from steam tables for dry, saturated steam (dryness fraction $q = 1$), and constant-pressure and constant-dryness curves are then added by solving the simultaneous equations—

$$\text{Total heat} = H = h + qL$$
$$\text{Entropy} = \varphi_{\,steam} = \varphi_{\,water} + (qL/T)$$

for as many values of pressure and dryness fraction as desired.

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* See page 551.

Curves for superheated steam are plotted by solving the simultaneous equations—

Total heat = $H + 0{\cdot}48\ (T_1 - T)$

Entropy = φ steam $+ 0{\cdot}48\ (\log_e T_1 - \log_e T)$.

STEAM ENGINE MECHANISM.

The mechanism of a simple steam engine may be represented as in Fig. 5. In order to determine the crank effort produced

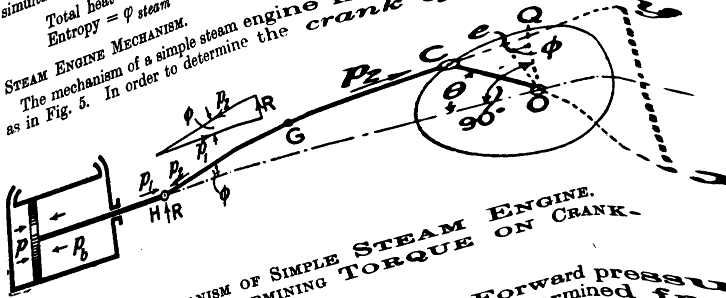

FIG. 5.—MECHANISM OF SIMPLE STEAM ENGINE. CONSTRUCTION FOR DETERMINING TORQUE ON CRANK-SHAFT.

at any moment the procedure is as follows. Forward pressu on piston p lbs. per sq. in., back pressure p_b (determined fr the indicator diagrams for left and right-hand ends of t cylinder respectively). Net pressure $P = p - p_b$.* Driv force on piston = PA lbs., where A = piston area in sq. From a piston acceleration diagram (see later) the pis acceleration a is determined for the piston position concer Then the accelerating force F required by the reciprocating F = Wa/g, where W is the weight of the piston, etc and g = acceleration due to gravity = 32·2 ft. per sec.². driving force available at the crosshead is PA − F, say p

At the crosshead H, p_1 is in equilibrium with the thr in the connecting rod and the reaction R of the crosshead The thrust p_2 acting at C produces a torque on the cran $= p_2 \times Oe$. From the vector triangle for the forces op at H, it is seen that $p_2 = R/\sin \varphi$. Again, in the OeQ, $Oe = OQ \cos \varphi = OH \sin \varphi$. Therefore $p_2 \times$ $\times$ OH. Also, since $p_2 = p_1/\cos \varphi$, $p_2 \times Oe = p_1 \times$ O words: the torque on the crankshaft equals the product of the force at the crosshead by the intercept obtained by producing the line of the connecting rod to cut the vertical through O.

---

* When expansion has proceeded so far that the driving is less than the back pressure p_b (which is then increasin compression on that side of the cylinder), the crank h the engine. The points A and B (Fig. 8), at which $p = p_b$ as the *change-points*. The reversal of stress which then o connecting rod, etc., is apt to cause knocking. A main f Willans' single-acting engine is the use of such air-cushioni is no change-point. In the Belliss engine there is a but knocking is prevented by forced lubrication.

This product may be obtained geometrically as follows: Produce the crank radius CO and set off $Cx = p_1$ to any convenient scale. Draw xy perpendicular to HO produced, then: Torque on crankshaft = Crank radius × xy (xy being read by the scale adopted for p_1).

To determine the accelerating force F (for deduction from PA), it is necessary to know the weight W and the acceleration a of the reciprocating parts; then $F = Wa/g$ where $g = 32{\cdot}2$ ft. per sec. per sec. The *weight of reciprocating parts* is the weight of the piston, piston rod and crosshead *plus* a fraction of the weight of the connecting rod. If G (Fig. 5) be the centre of gravity of the connecting rod—determined by balancing the rod on a knife edge—the weight to be added to that of the piston, etc., is total weight of connecting rod × GC/CH. This result may be determined directly by resting C on a suitable edge and H on a spring balance, the reading of which is then the desired weight.

Piston acceleration may be calculated from the formula—

$$a = \omega^2 r \left(\cos\theta + \frac{r}{l}\cos 2\theta\right),$$

where ω = angular velocity of crank, in radians per sec.
r = crank radius, in feet
θ = crank angle, in radians
l = length of connecting rod, in feet
a = piston acceleration, in feet per sec. per sec.

Bennett's construction for piston acceleration is very simple and accurate. Referring to Fig. 6, the connecting rod is first drawn in the position corresponding to crank angle 90° (or 270° in order to avoid overlapping of construction lines) and OR is dropped perpendicular to LP. Then $PL \times PR = PO^2$. To find the piston acceleration corresponding to any crank angle θ, R is marked on the corresponding position of the connecting rod, i.e. CR = PR. Draw RS perpendicular to CH; ST perpendicular to OH; and TU perpendicular to CH.

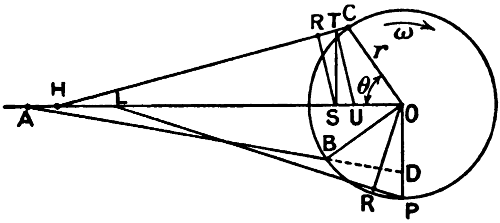

Fig. 6.—Graphic Determination of Piston Velocity and Acceleration.

Then UO represents the piston acceleration to the same scale as CO represents the radial acceleration $\omega^2 r$ of the crank pin. By repeating this construction for a number of values of θ the acceleration diagram for the complete stroke can be plotted. If the scale of ordinates be multiplied by W/g, the same diagram represents the force required for acceleration. If preferred, this force may be reduced to a value per sq. in. of piston and be then subtracted from the ordinates of the effective pressure diagram. Owing to the finite length of the connecting rod the acceleration diagram is not symmetrical about the centre of the stroke.

The graphical construction for determining *piston velocity* is very simple. Referring to Fig. 6, the piston velocity corresponding to any crank position OB is determined by producing the line AB of the connecting rod to cut the vertical through O in D, then—

$$\text{Piston velocity} = \text{Crank velocity} \times OD/OB$$

Crank Angle for Maximum Piston Velocity. During the greater part of the revolution the crank pin velocity is greater than the piston velocity, but the latter is the greater during tw portions of the revolution when the connecting rod is nearl tangential to the crank circle. Approximately, the pisto velocity is greatest when the crank and connecting rod are a right angles. Determination of the exact crank angle at whic the piston velocity is greatest is a matter of some difficult Unwin* gives the following extremely approximate and ve simple expression—

$$\sin^2\beta = \frac{n^2}{n^2+1} - \frac{2n^2+1}{(n^2+1)(n^4-4n^2)}$$

where β = angle between crank and line of stroke w piston velocity is a maximum;

n = ratio of length of connecting rod to crank rad

This expression gives very nearly exact values for β and a m closer approximation than the relation—

$$\tan\beta = n,$$

which gives the crank angle at which the crank and conn rod are at right angles.

Simple Single-acting Steam Engine and Indicator Di The action of steam in a single-acting engine may be ex by reference to Fig. 7. It is assumed that the stroke is co and that the valves are actuated as desired. The i diagram represents the pressure and volume of stea cylinder at any moment. The cylinder diameter being the volume of steam varies directly with the positi piston but *not* with its distance from the end of th because there is a certain volume, OA, of steam in th when the piston is at the end, 1, of its stroke. T OA is the clearance volume and may be up to 10 P the volume swept by the piston.

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* Proc. Inst. Civil Eng., Vol. CXXV, Part III.

At the beginning of the stroke the steam valve S is open and steam at constant pressure P is admitted until the piston reaches the point, 2, of *cut-off*. The steam valve is then closed and the steam in the cylinder expands, decreasing in pressure meanwhile until, at the *release point* 3, the exhaust valve E is opened and the pressure falls quickly to atmospheric pressure in the case of a non-condensing engine.* The timing of the valves is such that this pressure is reached when the piston reaches the end, 4, of its stroke. During the return stroke the exhaust valve remains open from 4 to 5, and the pressure in the cylinder remains constant at the *back pressure, p*. At the *compression point*, 5, the exhaust valve is closed and the steam trapped in the cylinder is compressed by the energy stored in the piston and connecting rod, etc., its pressure rising as shown by the curve 5–6. At the *admission point* 6 the steam valve is opened

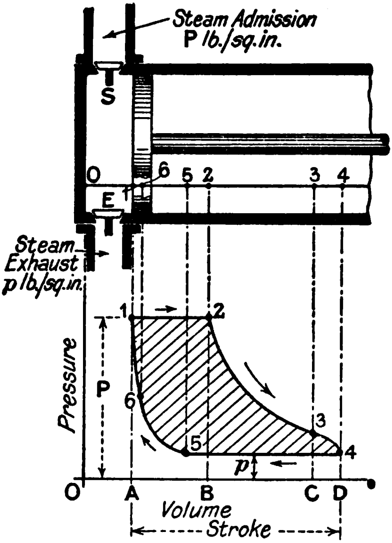

FIG. 7.—SINGLE-ACTING ENGINE AND INDICATOR CARD.

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* In this case the engine is taken to be non-condensing because atmospheric pressure exists behind the piston (unless the outer end of the cylinder be closed and connected to a condenser and air pump). If a single-acting engine worked condensing with the front of the cylinder open, the load would have to do work upon the piston as long as the pressure behind the latter was below atmospheric.

and the pressure in the cylinder rises rapidly to P. Thereafter the cycle is repeated.

In the above remarks no account is taken of pressure drop due to frictional resistance to flow through valves or steam passages.

The shaded area inside the indicator diagram is proportional to the external work done by the steam during the cycle. The earlier the cut-off, i.e. the smaller the volume of steam in the cylinder when the admission valve is closed, the smaller the area within the diagram or "card," the greater the expansion ratio (i.e. Volume at exhaust/Volume at cut-off = OD/OB—Fig. 7), and the lower the pressure at the point, 3, of exhaust. The lower the pressure at 3 the less energy is sacrificed by the unresisted pressure drop between 3 and 4 after the exhaust valve is opened but, as the area enclosed by the diagram is reduced, the work done per stroke is reduced, i.e. the power of the engine is decreased. With very early cut-off steam may be reduced to the back-pressure p before the point 3 is reached. Negative work is then done (i.e. the load drives the engine until the exhaust valve is opened and the pressure then rises by inflow of steam or air.

The point of cut-off is varied widely and is the means of varying the output of the engine. The points of release, com pression and admission are usually set once for all. Th timing of all these points is expressed by the percentage of th stroke at which they occur. When working at full power th cut-off may be at 30 per cent. of the stroke, measured from Fig. 7. The release point is usually at about 96 per cent. of t stroke, also measured from A. The compression point may at 85 per cent. and the admission at 95 per cent. of the retu stroke, measured from D.

DOUBLE-ACTING ENGINE.

In a double-acting engine the pressure on one side of piston is opposed by a certain back pressure on the other

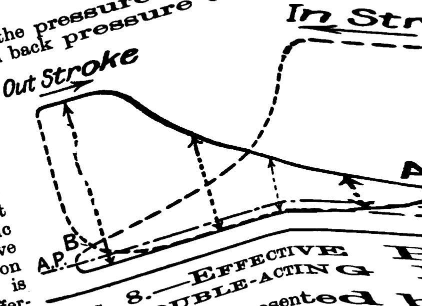

FIG. 8.—EFFECTIVE
IN DOUBLE-ACTING

the value of this pressure being determined from the indicator card for that end of the cylinder. We have now two cards to consider, as in Fig. 8 (which assumes that the engine works condensing so that the exhaust line is below atmospheric pressure). The effective pressure on the piston during the out stroke is represented by the difference between the solid portions of each card, and is negative beyond A, whilst the effective pressure on the in-stroke is represented b ence between the dotted portions of each card, an

In practice, with the valves set accurately t ond difference between the two cards is so small that, for ma purposes, it is sufficient to take a card from one end of t cylinder.

CLEARANCE VOLUME.

This is usually expressed as a percentage of the effecti volume, i.e. of the volume swept through by the piston. Whe applicable the most accurate method of measuring the clearanc volume is to fill the whole clearance space *right up to the stea* valve seating with water from a graduated cylinder. If th steam valve is not at the highest point in the clearance space if there are pockets in the latter, or if the above method i inapplicable for any other reason, the clearance volume may b determined from the drawings of the engine.

Typical average values for the clearance volume, as a percentage of the effective volume are: 5 per cent. in engines with Corliss valves; 10 per cent. with slide valves; 15 per cent. with piston valves.

INDICATED HORSE-POWER—BRAKE HORSE-POWER—MECHANICAL EFFICIENCY.

The *indicated horse-power* of an engine is the horse-power as determined from the indicator diagram. It is the horse-power developed at the piston of the engine and must be reduced by all the frictional losses of the engine and by the power absorbed in driving valve gear, etc., in order to determine the *brake horse-power* available at the crankshaft.

The ratio Brake h.p./Indicated h.p. = Mechanical efficiency of the engine and may be 90 to 95 per cent. in single cylinder condensing engines and 75 to 85 per cent. in condensing engines if the power taken to drive the air pump be included as part of the engine losses.

The power required to drive an engine when unloaded (determined from the "friction card" of the engine) is practically the same as that required to overcome friction in the engine when fully loaded. In the case of condensing engines, the power absorbed by the air pump increases with the load on the engine, but in all cases the actual friction of the engine is practically constant at all loads. This being so, the mechanical efficiency of the engine rises from zero on no-load (all power developed expended in overcoming friction) to a maximum at the maximum output of the engine. The mechanical efficiency of a Paxman-Lentz 600 h.p. cross-compound condensing engine, with double-beat valves (p. 489), is about 86 per cent. at half-load, 91½ per cent. at full-load; and 92 per cent. at 1½ times full-load. The mechanical efficiency of a Marshall 32 h.p. single cylinder non-condensing engine, with balanced piston valves (p. 499), is about 95½ per cent. at full-load.

The frictional loss in steam engines may be reduced by ensuring good lubrication (particularly in the cylinders, crosshead slides, and main bearings); by using long connecting

rods (to reduce load on crosshead slides) ;
packed piston rings and balanced slide valve
unbalanced pressure between sliding surfaces
and by using anti-friction gland packings pro
Thurston states that the distribution of fric
with balanced slide-valve is—

Main bearings . . . . .
Piston and rod . . . . .
Crank pin . . . . . .
Crosshead and gudgeon pin . .
Eccentric strap . . . . .
Valve and rod . . . . .

DETERMINING INDICATED HORSE-POWER.

Let the "mean effective pressure" on the
per sq. in. during a stroke of length *l* ft. ; the
be *a* sq. in. ; and the number of strokes per min.
if the engine be double-acting. Then the to
on the piston = *pa* lb. ; the work done per str
the work done per min. = *plan* ft. lb., and th

Indicated horse-power = *plan*/3

which formula is memorized easily.

The *mean effective pressure* (m.e.p.) is, stri
difference between the steam-line pressures
indicator diagram for one end of the cylinder
line pressures as indicated by the card for the
cylinder. In practice it is usually sufficie
assume that the diagrams for each end of
identical and to take as the m.e.p. the press
to the mean width of a single card.

Example. The card shown in Fig. 9 was
or 1/40 spring (i.e. 1 in. vertically or card rep
sq. in.). All that is available in the first insta
and the atmospheric line AP. If it is desir
pressures from the card the zero pressure line
case 14·7/40 = 0·37 in. below AP. The sca
then be set out as shown. To determine th
lines AA, BB are drawn through the ends of th
length of the latter is divided into 10 equal pa
easily by setting a centimetre scale across the
zero of the scale is on AA and the 10 cm. n
intermediate centimetres are marked on the
lines are drawn through these points as cl
widths w_1, w_2, and w_3, etc., of the diagram
at the centres of the ten sections and the su
is divided by 10 to determine the mean wi
found to be 0·87 in., i.e. the m.e.p. = 0·87
per sq. in.

The cylinder diameter is 18 ins. (piston area = 254 sq. in.), the stroke = 2 ft., and the engine speed = 100 r.p.m., i.e. 200 strokes per min. Hence the indicated horse-power = 34·8 × 2 × 254 × 200/33000 = 107 i.h.p. Allowing for possible errors in the indicator, for the assumptions made in determining the m.e.p., and for the approximate method used in determining the mean width of the diagram, the most we are justified in saying is that the engine develops *about* 100 *i.h.p.* Such statements as "the engine develops 1003·4 i.h.p." imply a degree of accuracy far in excess of that which can actually be obtained. The indicator itself is responsible for errors usually amounting to 3 per cent. or more and hardly reducible by any means below 1 per cent.

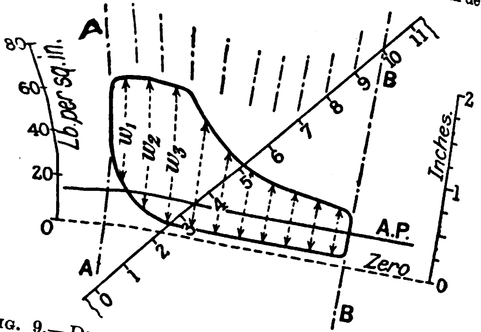

FIG. 9.—DETERMINING MEAN EFFECTIVE PRESSURE.

A *more accurate method* of determining m.e.p. is to measure the area enclosed by the indicator diagram (either by planimeter or by squared paper) and divide this by the length of the diagram.

DIAGRAM FACTOR.

This factor is defined as the ratio of the mean effective pressure determined from the actual indicator card to that determined from an ideal card constructed by neglecting clearance and compression, and assuming constant admission pressure, hyperbolic expansion, and instantaneous release.

Unwin states that the diagram factor in simple engines varies from 0·7 in engines with much compression to 0·8 or 0·9 in engines with moderate compression.

Conditions which reduce the diagram factor are high compression, wire-drawing, condensation, the presence of a negative loop in the card (due to operating at excessive ratio of expan-sion), and receiver losses in compound engines. Typical values of diagram factors are—

Simple slide valve engines, unjacketed	0·7 for high piston speeds
Simple slide valve engines, unjacketed	0·8 ,, low ,,
Corliss engines and slow speed slide valve engines, jacketed	0·85 to 0·95
Compound (receiver) and triple expansion engines, unjacketed	0·6 to 0·65
Compound (receiver) and triple expansion engines, jacketed	0·65 to 0·7
Tandem compound engines, jacketed	0·7 to 0·8.

INDICATORS.

The ordinary steam indicator consists essentially of a spri loaded piston in a cylinder the lower end of which is connect to the main engine cylinder through a small pipe and co The movement of the indicator piston is proportional to t pressure in the main cylinder and is recorded by a pencil mount on a magnifying mechanism driven by the indicator pist The pencil records on a chart which is clipped round a dru The latter is rotated (against the control of a spring) by the of a string which is coupled to the crosshead of the en through a suitable reducing mechanism. The record on indicator card is thus produced by two motions, one proporti to the pressure in the main cylinder and one proportional t corresponding position of the main piston. When the indi cock on the main cylinder is closed, the pencil traces the at pheric line on the card. Given the atmospheric line and k ing the strength of the indicator spring the card ca calibrated as explained above.

Springs of various strengths are used to suit various pressures. Very weak springs are used to obtain an er record of the exhaust line, etc., and a stop is then used t straining the indicator during the high-pressure period stroke.

Errors may be introduced by change in strength of cator springs which should therefore be calibrated fr This calibration should be carried out at working tem Error may also be introduced by throttling in long pipe engine and indicator; by friction or slackness in the mechanism; and by geometrical errors in the latte mechanism used to oscillate the paper drum. In are serious where high-speed engines are concerne optical indicators have been designed for prec Continuous indicator diagrams may be made phot or on a continuous band of paper. For detail indicators and their manipulation the reader is text-books and makers' descriptive pamphlets.

GEOMETRY OF INDICATOR CARDS.

(1) *Calibrating Pressure and Volume Axes.* Referring to Fig. 10, the card and the atmospheric line A.P. are all that are available at first. If the rating of the spring used be S lb. per in., set out a line AB, 14·7/S in. below and parallel to the A.P. line. This is the line of zero pressure. The length CD represents the effective cylinder volume V $= \pi d^2 l/4$ cu. ft. where d = piston diameter, ft.; and l = stroke, ft. Draw a line CE at any convenient angle and set out CV representing V to any convenient scale. Set out Cv representing the clearance volume and CF representing 1 cu. ft., to the same scale. Join VD and draw vv^1, Ff parallel to VD. Set out CO = Cv^1, then O is the zero of volume. Set out Of^1 = Cf representing 1 cu. ft. and subdivide Of^1 as required. The diagram is then calibrated and the absolute pressure and volume corresponding to any point can be read off at once.

(2) *Shape of Expansion Curve.* The expansion curve is represented very closely by an equation of the form $pv^n = C$, where p = absolute pressure, v = volume, n = numerical index, C = constant. With an engine in good running condition the expansion curve is almost exactly a *rectangular hyperbola*, and a curve of this standard form may be drawn through the cut-off point as follows: Drop verticals through the cut-off, A, and end of expansion B Fig. 11. Divide ab into any convenient number of equal parts. Draw rays Pb Pc, and lines Ab^1, Ac^1, parallel to them. The intersections between Ab^1, Ac^1, and the verticals through b, c, are points on the rectangular hyperbola. If the actual expansion curve is above the "standard" curve there is leakage through the valves; if below, leakage past the piston.

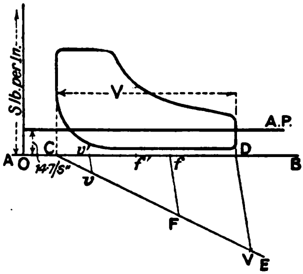

FIG. 10.—CALIBRATING INDICATOR CARD.

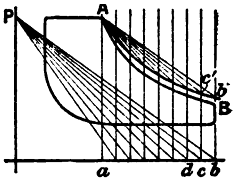

FIG. 11.—CONSTRUCTION FOR RECTANGULAR HYPERBOLA.

Assuming the equation for the actual expansion curve to be $pv^n = C$ the value of n and C may be found by tabulating corresponding values of p and v, and plotting log p against log v. The slope of the straight line thus obtained $= n$ (the scale value of ordinate/abscissa being used and not the geometrical ratio if different scale units be employed). The intercept on the axis of log p = log C, whence C is determined by antilogarithms.

A graphical construction for a curve of the form $pv^n = C$ is illustrated by Fig. 12. The equation for the *adiabatic curve*

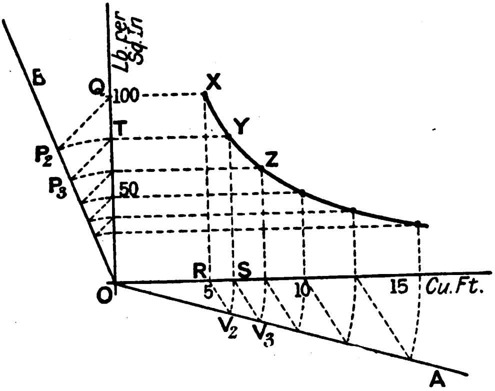

FIG. 12.—GRAPHICAL CONSTRUCTION FOR ADIABATIC $PV^n = C$.

for steam is approximately $pv^{1\cdot135} = C$. Suppose that we require the adiabatic through X (Fig. 12) for which $P_1 = 100$ lbs. per sq. in. and $V_1 = 5$ cu. ft. We have $P_1 V_1^n = P_2 V_2^n$. Taking $V_2 = \frac{5}{4} V_1$ and $n = 1\cdot135$ (for d.s. steam),* we have $V_2 = 6\frac{1}{4}$ cu. ft., and $P_2 = 100\ (4/5)^{1\cdot135} = 77\cdot63$. Draw the lines OA, OB at any convenient angles and set out along them, to scale, V_2 and P_2. With centre O and these radii transfer V_2 to S and P_2 to T. The intersection of the vertical through S with the horizontal through T gives a second point Y on the adiabatic. Through T draw TP_3 parallel to QP_2; through S draw SV_3 parallel to RV_2. Repeat the above construction to obtain Z, and so on.

It is more accurate and not much more trouble to plot the *isothermal* (constant temperature) $pv = C$ by calculation than by the graphical method. On the other hand, the graphical method saves troublesome calculation where the *adiabatic*

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* Zeuner's empirical formula for steam is: $n = 1\cdot035 + 0\cdot1q$ where q = dryness fraction at cut-off.

(constant heat) curve $pv^n = C$ is concerned. The adiabatic equation for dry saturated steam is often convenient in the form $pv^{17/16} = 479$, where p = absolute pressure in lbs. per sq. in., and v = cu. ft. per lb. of d.s. steam.

Area under Isothermal and Adiabatic Curves. If P_1, V_1 represent the pressure and volume at one point on an expansion curve and P_2, V_2 the pressure and volume after expanding to a second point on that curve, the area under the curve between the two points may be calculated as follows : *Isothermal*—The law of the curve is $PV = C$. The area under the curve $= PV \log_e r$, where $r = V_2/V_1$. *Adiabatic*—The law of the curve is $PV^n = C$. The area under the curve is $(P_1V_1 - P_2V_2)/(n - 1)$, where $n = 1{\cdot}035 + 0{\cdot}1q$, and q = initial dryness fraction of the steam considered.

USE OF INDICATOR DIAGRAMS.

Indicator diagrams may be used for many purposes, notably for determining: (1) Indicated horse-power. (2) Cylinder dimensions required for stated horse-power. (3) Steam consumption per i.h.p.-hour. (4) Thermal efficiency of engine. (5) Weight of steam shut in at compression. (6) Dryness fraction of steam during expansion. (7) Investigation of losses in valves, leakage past piston, etc. The accuracy of these various determinations is limited by the accuracy of the indicator itself and by the assumptions made in utilizing the card. The nature and degree of these limitations should be considered in each particular case otherwise very misleading results may be obtained.

The determination of *indicated horse-power* has already been considered. Other applications of indicator diagrams are discussed below.

Cylinder Dimensions for Stated Horse-power. A diagram is first plotted showing the behaviour of 1 lb. of steam under the conditions of the design. The volume of 1 lb. of steam, at the admission pressure and at whatever dryness fraction is assumed, is set off as at AB, Fig. 13. The expansion curve BH is then

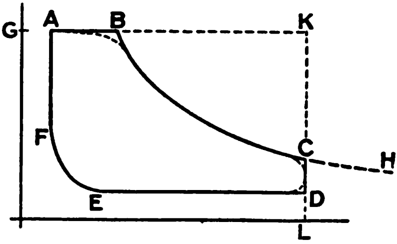

FIG. 13.—CONSTRUCTION FOR DETERMINING CYLINDER DIMENSIONS.

drawn. Assuming this to be a rectangular hyperbola it may be constructed as in Fig. 11. If cut-off is to be at 25 per cent, AK is made = 4 × AB and the perpendicular KL then cuts BH at the release point C. The back-pressure line DE is then set off*, and through the compression point E (at say 85 per cent of the stroke), a rectangular hyperbola EF is drawn for the compression line. AG = clearance and may be 10 per cent for a slide-valve engine. The actual card would depart considerably from this theoretical form and would be subject to wiredrawing, etc.

The area of the diagram being measured and the m.e.p. thus determined, it may be found that the card would have to be repeated n times per min. to yield the desired i.h.p. Then if the desired engine speed corresponds to N strokes per min., it is obvious that the cylinder feed per stroke must be 1 lb. $\times n/N$. Knowing the admission pressure the corresponding volume of steam can be determined and thence the cylinder diameter, the stroke being fixed.

Determining Steam Consumption from Indicator Card. (1) For approximate results neglect the clearance space and assume the steam in the cylinder to be dry and saturated. From a calibrated indicator card determine the volume from admission to cut-off. Suppose that this is v cu. ft. at a pressure p lb./sq. in., also determined from the card. From steam tables the weight of dry saturated steam at p lb./sq. in. is found to be V cu. ft. per lb., hence the weight of steam in the cylinder (on the above assumptions) is v/V lb. and the consumption per i.h.p.-hr. is $60\ nv/PV$; where P = i.h.p. of engine, and n = number of strokes per min.

Thermal Efficiency of Engine. Suppose that the i.h.p., determined from the indicator card, is P. Then work is done by the engine at the rate of 550 P ft.-lb./sec. or 550 P/778 i.e. 0·71P B.Th.U./sec. = H_w B.Th.U./sec. The weight of steam consumed per second is determined by aid of steam tables as already explained. The total heat H_s in this steam, above hot-well temperature, is $H + (t - 32)$ where H is the total heat *of the weight of steam concerned* as determined from the total heat per lb. given in steam tables, and t = hot well temperature, °F. Then thermal efficiency of engine = $100\ H_w/H_s$ per cent.

Weight of Steam Shut in at Compression. Assuming the steam to be dry and saturated during compression, the weight = volume as determined from card/tabular volume (cu. ft. per lb.) at the pressure corresponding to the point selected on the card. If the steam has dryness fraction q the "dry, saturated" volume, V^1, corresponding to the volume, V, as determined from the card is V/q, and the weight of steam = $V/(q \times$ tabular volume). This expression neglects the volume occupied by the water in wet steam (0·016 cu. ft. per lb. of water).

Dryness Fraction of Steam during Expansion. If the weight of steam corresponding to a point A on the expansion curve (Fig. 14) were determined from the expression v/tabular volume, the result would be far from the truth because the steam is far from being dry and saturated. By measuring the boiler feed per hr. or (preferably) the weight of condensate per hr., and dividing by the number of engine strokes per hr., we obtain the mean weight w of cylinder feed per stroke (the engine being on constant load during the test period). If the steam shut in at compression (see previous paragraph) be w_1 lb., the total weight of steam expanding is $W = (w + w_1)$ lb. The tabular volume at pressure p being V_t, the volume of W lb. of d.s. steam is $WV_t = V$ cu. ft., but the actual volume is v cu. ft., hence the dryness fraction is $v/V = pA/pV$, Fig. 14. If the saturation curve corresponding to the theoretical volumes during expansion (d.s. steam throughout) be plotted as shown, the dryness fraction corresponding to any point B is AB/AC.* The departure of the expansion curve from the saturation curve is due partly to wetness of the steam and partly to leakage through valves.

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* Say 2 to 3 lbs. per sq. in. above atmosphere for a non-condensing engine, and 5 to 7 lbs. per sq. in. absolute for a condensing engine.

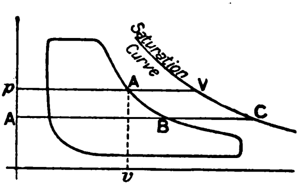

FIG. 14.—DIAGRAM ILLUSTRATING DRYNESS FRACTION OF STEAM DURING EXPANSION; VOLUME BASIS.

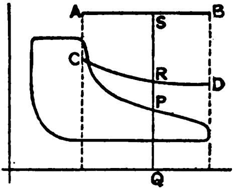

FIG. 15.—DIAGRAM ILLUSTRATING DRYNESS FRACTION OF STEAM DURING EXPANSION; WEIGHT BASIS.

Instead of determining the dryness fraction from the ratio of the actual to the tabular volume of the weight W of steam, we may calculate the weight w_c corresponding to any point on the expansion curve, assuming the steam to be dry and saturated. The weight thus calculated will be less than the actual weight W ($= w + w_1$) and $q = w_c/W$, if there be no leakage of steam. Referring to Fig. 15, the actual weight of steam expanding is constant and may be represented by AB. The weight w_c, calculated from the card, is variable and the dryness fraction corresponding to any point P on the expansion curve $= w_c/W = QR/QS$.

The dryness fraction at release is greater than at cut-off if the heat received by the steam from the jacket exceeds the heat converted to work during the stroke.

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* Labour is saved by first changing the volume scale of the diagram so as to represent the volume of 1 lb. of the cylinder charge under the conditions represented by the diagram.

DIAGNOSIS OF DEFECTS FROM INDICATOR CARD.

(*a*) *Leakage at Piston or Valves.* If the expansion curve falls much below the rectangular hyperbola through the point of cut-off (see Fig. 11) there is probably leakage past the piston rings ; whilst if the curve rises above the rectangular hyperbola, there is leakage at the valves.

(*b*) *Wiredrawing* is the name given to the pressure drop caused by the resistance offered by valve ports and steam passages to the flow of steam. The admission line in the high-pressure card of a compound locomotive engine, for instance, may be practically continuous with the expansion curve, as at ADF, Fig. 16, instead of being horizontal as far as the cut-off C, and then expanding along the line CE. The area ACD represents work done in getting the steam into the cylinder and there is a further loss owing to the fact that the expansion curve commences at D, i.e. at a lower pressure than C. Wiredrawing is due mainly to the gradual closing of the admission port by the slide valve when the piston is moving at or near its maximum speed, and is not produced to anything like the same extent where Corliss or drop valves are used, these closing practically instantaneously, i.e. remaining full-open until the last moment.

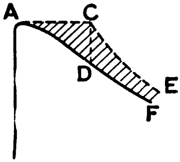

FIG. 16.—LOSS DUE TO WIRE-DRAWING.

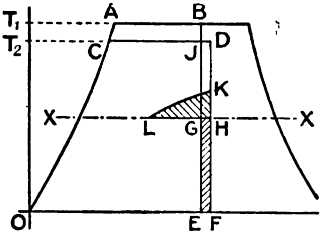

FIG. 17.—LOSS OF EFFICIENCY DUE TO WIREDRAWING AND PRESSURE-DROP AT RELEASE.

"Wiredrawing" results in a pressure drop of steam without external work being done. The total heat, therefore, remains the same and, the total heat of dry saturated steam being lower at lower pressure, the steam is dried to some extent by wiredrawing. Referring to Fig. 17, the total heat in the steam at state-point B is represented by OABE. This total heat being the same in the wiredrawn steam at lower temperature T_2, the state-point must be at D such that OCDF = OABE. The point D corresponds to a higher dryness fraction than at B. The heat represented by CABJ equals that represented by JDFE, but only JDHG (above the exhaust line XX) is recoverable. The heat represented by GHFE is lost to the cycle.

(*c*) *Pressure Drop at Release.* The sudden pressure drop at release is accompanied by no further movement of the piston and therefore represents a loss of energy. Referring to the entropy diagram, Fig. 17, instead of expanding adiabatically from D to H, the steam is released at a higher pressure represented by K and its state-point moves along the constant volume line KL to the exhaust line XX. The heat represented by the area KHL is lost to the cycle. With a simple slide valve earlier cut off involves earlier release and compression. To reduce the wasteful pressure drop at release, stationary engines usually cut off at $\frac{5}{8}$ – $\frac{7}{8}$ of stroke, $\frac{3}{4}$ being average. By using variable expansion valves the advantage of early cut-off and expansive working can be obtained without correspondingly early release and excessive compression.

COMPOUNDING.

The principle of compound engines is simply that the expansion from h.p. admission to l.p. exhaust is conducted in two or more cylinders in order to reduce the temperature range of expansion in each cylinder, and thus to reduce initial condensation. It is found that the saving effected by reduced condensation in the cylinders exceeds the loss introduced by condensation in the pipes and receivers between cylinders. Referring to Fig. 18, if steam were expanded from 120 to 12 lbs. absol. in a single cylinder (as shown by the heavy diagram), the temperature of the steam would fall from 341° F. to 202° F. (a range of 139° F.), and the cylinder wall would be much cooler at release than at cut-off. If the expansion were split between two cylinders as indicated there would be certain losses (shown by the shaded area), but the temperature range in each cylinder would be only about 70° F. and the point X (corresponding, say, to a dryness of 0·5 when working with a single cylinder), would move inwards to a point X_1.

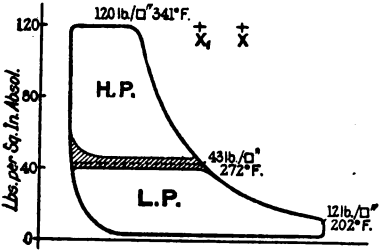

FIG. 18.—ILLUSTRATING THE PRINCIPLES OF COMPOUNDING.

The reduction in feed to the h.p. cylinder may be 20 or 30 per cent less than the feed to a single cylinder for the same total power.

The *power of a compound engine* is determined by the volume of the l.p. cylinder. The total ratio of expansion (on which depends the total power) = Vol. l.p. cylinder/Vol. of steam at h.p. cut-off. This ratio is independent of the pressure at which steam is transferred from h.p. to l.p. cylinder, and of the number of stages in the compounding. Due to elimination of transfer losses greater total power would probably be obtained by conducting the whole expansion in the l.p. cylinder but the steam consumption/i.h.p.-hr. would be higher, owing to increased total condensation.

Simple compound engines expand the steam in two stages, two l.p. cylinders being used if the dimensions of a single l.p. cylinder would be unwieldy, or if a 3-crank drive is required. *Triple expansion* engines expand the steam in three stages, the l.p. steam again being used in two l.p. cylinders if desired.

Quadruple expansion engines expand the steam in four stages.

The *ratio of cylinder diameters* is commonly 3 : 5 to 6 in compound engines; 3 : 4 to 5 : 7 to 8 in triple expansion engines; and 3 : 3⅓ to 4 : 5 to 6 : 7 to 10 in quadruple expansion engines. These ratios correspond to *ratios of cylinder volumes:* 1 : 2¾ to 4 in compound engines; 1 : 1¾ to 2¾ : 5½ to 7 in triple expansion engines; and 1 : 1¼ to 1¾ : 2¾ to 4 : 5½ to 11 in quadruple expansion engines.

SIMPLE COMPOUND ENGINES.

The simplest arrangement of cylinders is shown at A, Fig. 19. The arrangement, shown at B, of two h.p. cylinders exhausting into one l.p. cylinder has some advantages in relation to the balancing of the engine but it is more rational to split the l.p. cylinder into two as at C, Fig. 19. The advantage of this arrangement is that live steam can be introduced at P, between h.p. and l.p. cylinders, when the engine is under very high load; the control thus secured over the pressure in the l.p. cylinders is an important practical advantage.

TRIPLE EXPANSION ENGINES.

The simplest arrangement of cylinders is as at A, Fig. 20. If a single l.p. cylinder is inconveniently large the low-pressure stage may be conducted in two l.p. cylinders mounted side by side and both taking steam from the same pipe or receiver. The balancing of the engine is improved by mounting the two l.p. cylinders at opposite ends of the crankshaft, as at B, Fig. 20, the four cranks being at 90° or thereabouts, according to the system of balancing; some modification of the crank angles may be made to replace, more or less completely, the balance weights otherwise required. Approximately equal power is developed in the h.p., i.p., and l.p. stages, hence each l.p. cylinder develops about one-sixth of the total power and the l.p. pistons may be considerably lighter than the h.p. and i.p. pistons. The "natural" weights of the components of such an engine, as designed on the basis of strength alone, give nearly perfect balancing and the balance weights required are thus reduced to a minimum.

Indicator Cards for Compound Engines. The diagrams drawn by the indicator are closely similar for both the h.p. and l.p. cylinders but the length of the diagram represents, say, three times as great volume on the l.p. card as on the h.p. card and the strength of spring used when taking the h.p. card is probably about three times as strong as the spring used for the l.p. card. The total h.p. of the engine may be obtained by determining the h.p. of each cylinder from the appropriate card and adding the values obtained. In order to investigate the expansion as a whole, and the losses between cylinders, the high-pressure card (and intermediate pressure card if any) are replotted on the same sheet and to the same scale as the l.p. diagram. If more convenient all the cards can be replotted to some other scale.

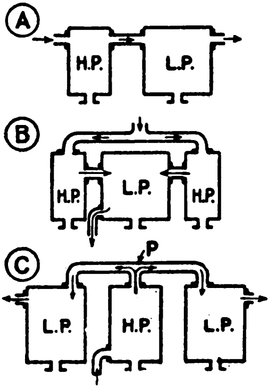

FIG. 19.—DIAGRAMMATIC REPRESENTATION OF CYLINDER ARRANGEMENTS IN COMPOUND ENGINES.

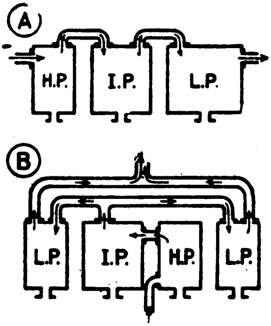

FIG. 20.—DIAGRAMMATIC REPRESENTATION OF CYLINDER ARRANGEMENTS IN TRIPLE EXPANSION ENGINES.

When the cards are thus brought together on a common scale the space between them represents loss in the receiver. The actual form of the exhaust line for the h.p. cylinder is determined by the relative timing of the h.p. and l.p. valves, and by the relative volumes of the cylinders and of the receiver. If the l.p. admission occurs after the commencement of h.p. exhaust, the exhaust line of the h.p. cylinder rises for a time owing to accumulation of steam in the receiver, until the l.p. admission occurs. The pressure in the receiver then falls, hence the admission line of the l.p. card slopes downwards. The clearance volume is greater and the weight of steam expanding is less in the l.p. cylinder than in the h.p. cylinder. The saturation curves for the two (or more) cylinders are, therefore, not quite continuous.

The point of cut-off of the l.p. card moves on the expansion curve (instead of on a constant pressure line as in a simple engine) hence earlier cut-off in the l.p. cylinder of a compound engine increases the horse-power developed by the l.p. cylinder (the h.p. cylinder cut-off being the same in both cases).

If the l.p. card overlaps the h.p. card when both are plotted to the same scale heat has entered the steam between the two cylinders (e.g. as the steam passes through a pipe in the smoke box in the case of compound loco. engines).

ADVANTAGES OF SUPERHEATED STEAM.

With the development of improved packings for piston and valve rods, and improved lubricants for cylinders, the difficulties formerly experienced in operating engines with superheated steam have now been overcome. The reduction in steam consumption where superheated steam is used is attributable to the reduced condensation in the cylinders; there is *theoretically* no advantage in superheating, but the practical advantage is great. The following data give convincing proof of the economy to be derived from the use of superheated steam. It must be remembered, however, that there are circumstances under which the saving effected in the engine by the use of superheated steam is more or less completely offset by reduction in the boiler efficiency. The overall efficiency of boiler and engine should be considered where superheated steam is employed.

A triple expansion Corliss valve engine rated at 1,000 i.h.p., 80 r.p.m. when operating with steam at 180 lbs. and jet condensing, consumed 12·96 lbs. steam per i.h.p.-hr. at 194 lbs. without superheat, and 9·94 lbs. per i.h.p.-hr. at 200 lbs. when superheated 162° F. (Hick, Hargreaves' superheaters). Saving due to superheating 23·3 per cent.

A compound Corliss engine (Frikart system), yielding about 750 i.h.p. when working with steam at 81 lbs. per sq. in., gave the following test data—

Superheat at engine, ° F.	Nil	76·7	152·2	180·6
Coal per i.h.p.-hr., lbs.	2·17	1·99	1·94	1·64
Steam per i.h.p.-hr., lbs.	16·77	15·02	14·40	13·53
Saving of coal, per cent	—	8·15	10·59	24·41
Saving of steam, per cent	—	10·43	14·1	19·3

(From Report of the Association of Steam Users in Alsace, 1895.)

Tests on a 300 h.p. Belliss triple expansion engine operating at full load with steam at 155 lbs. per sq. in. and 26 in. vacuum gave the following results—

Superheat at engine, °F.	Nil	100	200	300
Heat per b.h.p.-hr., B.Th.U.	18,450	16,250	14,700	13,600
Steam per b.h.p.-hr., lbs.	16·7	14·2	12·2	10·8
Saving of heat, per cent	—	12·0	20·0	26·5
Saving of steam, per cent.	—	15·0	27·0	35·0

Tests on the s.s. *Demodocus** gave the following service data—

	Average for 3 years Saturated Steam.	Average for 2 years Superheated Steam.
Displacement, tons	12,000	12,265
Speed, knots	12·63	12·55
I.h.p.	3,896	4,090
Coal per i.h.p.-hr., lbs.	1·7	1·37

The engines in this vessel are tr. exp., 30″, 50″, 83″ × 60″ stroke; smoke tube superheaters are used, and the average saving in coal is 13·8 per cent per day (19·4% per i.h.p.–hr.).

RELATIVE EFFICIENCY.

Mechanical energy can be derived from heat only by allowing the latter to change from one temperature to a lower one. A heat unit at atmospheric temperature is useless as regards power development. Indeed, in reciprocating engines, it is not economical in practice to continue the expansion of steam after the latter has reached about 100° F.

From the *second law of thermo-dynamics*, if a quantity, Q, of heat be admitted to any heat engine at absolute temperature T_1 and exhausted at absolute temperature T_2, the amount of heat which can theoretically be converted to mechanical energy is $Q(T_1 - T_2)/T_1$. Thus if an engine receives steam at 366° F., i.e. 826° F. absolute, corresponding to a gauge pressure of about 150 lbs. per sq. in., and exhausts at a temperature of 100° F. (560° absolute) corresponding to a vacuum of 28 in. of mercury only (826–560)/826, i.e. 32·2 per cent of the heat input could be converted into mechanical energy, even by an engine entirely free from losses. An actual engine may convert 20 per cent of the heat received into mechanical energy and will then have an *efficiency ratio* or *relative efficiency* of 20/32·2 = 62 per cent. In other words, the engine yields 62 per cent of the mechanical energy which could be yielded by a perfect engine free from all losses working *between the same temperature limits*. Even the perfect engine could only attain 100 per cent efficiency when $T_2 = 0$, i.e. when the lower temperature limit is absolute zero.

In practice heat is not given to an engine at constant temperature T_1, but the temperature of rejection T_2 is constant in any particular case. The formula for *theoretical thermal efficiency* then becomes:

$$\text{Theoretical Thermal Efficiency} = 100\int \frac{T_1 - T_2}{T_1}\, dT_1 \text{ per cent.}$$

RANKINE CYCLE.

In the Rankine standard cycle the standard engine is supposed to receive 1 lb. of d.s. steam which is expanded adiabatically from the admission pressure of the actual engine to the back pressure of the latter, then exhausting at this pressure and warming the feed water to the condenser temperature. In the standard cycle there is no clearance, no wiredrawing, and no pressure drop at release.

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* J. Neill, Greenock Assocn. of Engineers, Dec., 1920.

Referring to Fig. 21, the narrow (solid black) rectangle *a* represents 1 lb. of water. This is converted to a volume AB of d.s. steam at pressure P by the expenditure of H heat units (see Steam Tables) and the external work done is $(P - p_b) \times 144 \times V/778$ B.Th.U. $= w$ B.Th.U. If the steam were exhausted from B, the work done would be represented by ABED and the thermal efficiency would be $100 \times w/H$ per cent.

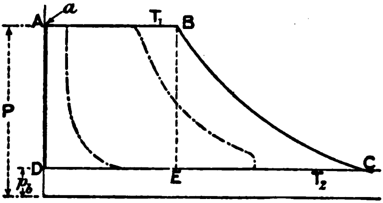

FIG. 21.—DIAGRAMMATIC REPRESENTATION OF RANKINE CYCLE.

Actually, however, the steam is expanded adiabatically from B down to the back pressure at C and the work done is represented by the area ABCD. It is more convenient to base calculations upon the initial and final absolute temperatures T_1 T_2 of the steam. Then the available work U B.Th.U. is—

$$U = (T_1 - T_2)\left(1 + \frac{L_1}{T_1}\right) - T_2 \log_e \frac{T_1}{T_2}$$

Feed being at T_2 and steam at T_1, the heat given to the steam is $L_1 + (h_1 - h_2)$ or, approximately, $L_1 + (T_1 - T_2)$ and the—

$$\text{Thermal Efficiency of Standard Engine} = \frac{U}{L_1 + (h_1 - h_2)}$$

[*Note.* If the Rankine cycle be considered for wet steam, it is necessary to read qL_1 instead of L_1 above.]

Having determined the "standard efficiency," the—

Relative Efficiency of Actual Engine

$$= \frac{\text{Actual Thermal Efficiency}}{\text{Thermal Efficiency for Standard Cycle}}$$

Entropy Diagram for Rankine Cycle. The events of the Rankine cycle are as follows: (1) 1 lb. of water at T_2 is heated to T_1, absorbing $(h_1 - h_2)$ B.Th.U. (2) 1 lb. of water at T_1 is converted to dry, saturated steam at T_1, absorbing L_1 B.Th.U. (3) The steam is expanded adiabatically from T_1 to T_2. (4) The wet steam at T_2 (dryness fraction q_2) is compressed isothermally, yielding 1 lb. of water at T_2 and rejecting q_2 L_2 B.Th.U.

On the entropy diagram, Fig. 22, these stages are represented by AB, BC, CD, and DA respectively. The heat absorbed in stage (1) is represented by EABF; and in stage (2) by FBCG. Stage (3) is adiabatic and in stage (4) heat represented by GDAE is rejected. Hence in a perfect engine the work done during the cycle is represented by the area ABCD.

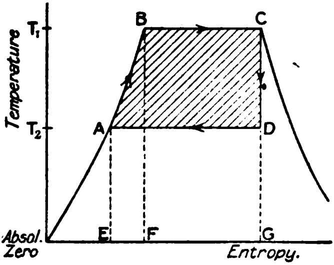

FIG. 22.—ENTROPY DIAGRAM FOR RANKINE CYCLE (NOT TO SCALE).

The *thermal efficiency of the cycle* = Area ABCD/Area EABCG, which ratio may be calculated from the formula given above.

ACTUAL THERMAL EFFICIENCY OF STEAM ENGINE PLANT.

If heat supplied to an engine were entirely convertible to mechanical work, the heat required per h.p.-min. would be 33,000/778 = 42·4 B.Th.U. As explained above, only about 20 per cent of the thermal input to a steam engine can be developed as mechanical energy even in favourable cases (the rest of the heat being rejected with the exhaust steam). The theoretical heat consumption of 42·4 B.Th.U. per h.p.-min. therefore becomes 42·4/0·2 = 212 B.Th.U. per min. per h.p. developed at the crankshaft (cf. Table V). Allowing for the fact that not more than 85 per cent (and usually less) of the thermal value of the fuel burned under the boiler is in the steam entering the cylinder (the remainder being lost up the chimney and by radiation, etc.); we have 212/0·85 = 250 B.Th.U. *at the grate* per h.p.-min. developed at the crankshaft, i.e. an overall thermal efficiency of 42·4/250 = 17 per cent as about the best result obtainable from a steam boiler and reciprocating engine combination.

For general estimating purposes the data in Table I, shown on page 459, are useful.

In the very best steam engines not more than 25 to 30 per cent of the thermal input can be converted to mechanical energy owing to limitations imposed by the atmospheric temperature. Complete conversion of thermal to mechanical energy would only be possible were the atmospheric temperature zero absolute (– 273° C.).

Thermal Efficiency in B.Th.U. per H.P.-Minute. It is usual to express the efficiency of a reciprocating steam engine in terms of the steam consumption per i.h.p.-hr. Actually the steam may be wet, dry and saturated, or superheated and in any case the total heat varies with the temperature and pressure

TABLE I.—AVERAGE THERMAL EFFICIENCY DATA FOR STEAM ENGINES.

	Thermal efficiency of engine at full-load.	Thermal efficiency of boiler.	Overall Thermal efficiency fuel to crankshaft.	B.Th.U. at grate per b.h.p.-min.	Coal (13,500 B.Th.U. per lb.) per b.h.p.-hr.
	Per cent.	Per cent.	Per cent.		Lb.
Simple steam engines . .	10 to 15	70 to 80	7 to 12	600 to 355	2·7 to 1·6
Compound, condensing . .	16 to 18	75 to 80	12 to 14½	355 to 295	1·6 to 1·3
Triple or quadruple expansion, condensing	20 to 24	75 to 80	15 to 19¼	284 to 220	1·25 to 0·98

of the steam. It is therefore impossible directly to compare the thermal efficiencies of engines for which the steam consumptions are known in lbs. per i.h.p.-hr. A definite basis on which to state and compare the engine efficiency is the heat consumption in B.Th.U. per i.h.p.-min.

In a report adopted by the Institute of Civil Engineers in 1898* it is recommended that: (1) "Thermal efficiency" of any heat engine = (Heat utilized as work on the piston) ÷ (Heat supplied to the engine). (2) Heat utilized be obtained by measuring indicator diagrams. (3) In the case of a steam engine, the heat supplied be calculated as the total heat of the steam entering the engine less the water heat of the same weight of water at the temperature of the engine exhaust, both quantities being reckoned from 32° F. (4) Temperature and pressure limits: *Upper limit,* the temperature and pressure close to, but on the boiler side of, the engine stop-valve.† In the case of saturated steam the temperature corresponding to the pressure can be taken. *Lower limit,* the temperature in the exhaust pipe close to, but outside, the engine. The temperature corresponding to the pressure of the exhaust steam can be taken. (5) The standard steam engine of comparison is the ideal steam engine working on the Rankine cycle between the same temperature and pressure limits as the actual engine. (6) The "efficiency ratio" = (Thermal efficiency of actual engine) ÷ (Thermal efficiency of corresponding standard engine). (7) The thermal economy of a steam engine should be stated in terms of thermal units required per min. per i.h.p. (and per b.h.p. if possible). (8) For scientific purposes there should also be stated the thermal units required per min. per h.p. by the standard engine of comparison. Reference should be made to the above-mentioned Report for a detailed consideration of the problem and numerical data and examples.

The Report of the Committee of the Institute of Civil Engineers on Tabulating Results of Steam Engine and Boiler Trials‡ is a valuable document containing a vast amount of information relating to engine and boiler efficiency factors and the methods of carrying out and interpreting trials.

TYPICAL STEAM CONSUMPTION, ETC.

Davey‖ gives much valuable information on the economy of condensing engines and includes references to a number of important engine trials. A table of trial data is included which may be summarized as in Table II.

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* *Report on the Definition of a Standard of Thermal Efficiency for Steam Engines* (Wm. Clowes & Sons, Ltd., 1s. net).

† Except that when the stop valve is purposely used for reducing the pressure the temperature of the steam at the reduced pressure is to be used for calculating the standard of comparison.

‡ Wm. Clowes & Sons, Ltd., 2s. net.

‖ *Proc. Inst. Civil Eng.*, Vol. CXXII, Part IV.

TABLE II.—TRIAL DATA FOR CONDENSING ENGINES.

Type of engine.	No. of engines.	Averages for No. of engines stated in Col. 2.				Min. and max. lbs. steam per i.h.p.-hr. for engines tested.	
		Initial pressure lb. per sq. in. absol.	Ratio of expansion.	Lbs. of saturated steam per i.h.p.-hour.	Units of work per unit of heat.	Min.	Max.
Cornish	5	33·65	3·65	21·00	80·2	18·82	24·00
Single-cylinder rotative .	4	76·67	3·95	23·51	76·44	19·15	32·14
Compound	10	84·3	7·73	18·53	92·11	15·6	21·73
Triple-expansion . . .	9	138·3	14·51	14·88	114·55	12·16	19·83

D. A. Low states* that Table III gives the results of tests of various types of condensing engines (compound, triple, and quadruple expansion) compiled from reliable sources.

TABLE III.—TYPICAL DATA FOR CONDENSING ENGINES (LOW).

Type of engine.	Initial pressure lb. per sq. in. absol.	Super-heat ° C.	I.h.p.	R.p.m.	Lbs. steam per i.h.p.-hr.	Heat supplied per i.h.p.-min. C.H.U.†	Thermal efficiency per cent.	Efficiency ratio per cent.
Stationary, compound	164	50	1208	100	12·27	136	17·3	56·0
	185	65	1400	80	11·74	129	18·3	64·5
	161	95	1234	100	10·75	126	18·7	59·0
	169	56	982	65	11·10	120	19·6	70·0
	129	213	258	101	8·58	107	22·1	67·0
Marine, triple expansion	180	nil	645	61	13·35	139	17·1	59·0
	160	nil	1994	72	14·98	148	16·1	64·0
Marine, quadruple expansion	214	nil	9099	78	13·47	138	17·2	63·0

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* In his excellent treatise, *Heat Engines* (Longmans Green).

† 1 C.H.U. = $\frac{9}{5}$ B.Th.U.

To these figures may be added those in Table IV from various sources.

Table IV.—Further Typical Data for Condensing Engines.

Type of engine (all condensing).	Initial pressure lb. per sq. in. absol.	Super-heat ° C.	I.h.p.	R.p.m.	Lbs. steam per i.h.p.-hr. at full load.
Uniflow . . .	195	110	500–1,500	—	10·5–10·0
Horiz. cross, compound Corliss .	150	nil	2,000	55	13·5
Tr. exp., vertical Corliss . .	210	nil	1,000	80	13·0
Ditto . . .	215	90	1,000	80	9·9
Tr. exp., Belliss Morcom. . .	173	72	620	350	11·9
Horiz. compound Paxman-Lentz .	170	nil	200	210	13·25
	150	83	250	200	10·4
	170	nil	366	190	12·3
	170	83	366	190	10·4
	150	97	570	155	10·3
	170	55½	600	155	10·9
	170	83	600	155	10·6

Table V (page 464), based upon one in *Hütte*, gives useful data concerning the steam consumption and thermal efficiency attainable in very good steam engines of various types.

Function of Valves and Valve Gear.

The sole function of the *valves* in a steam engine is to admit to, cut off, release and enclose steam in the cylinders at appropriate moments, the timing of the various events being controlled by the setting of the *valve gear* which operates the valves. Independent valves may be used for the several purposes or steam flow through and between various openings may be controlled by a valve moving over the latter.

Valve Terminology.

If a slide valve be placed centrally over the ports the amount by which the valve overlaps the steam port on the admission side is the *steam lap*, and the amount by which the valve overlaps the steam port on the exhaust side is the *exhaust lap*. Locomotive slide valves have frequently a negative exhaust lap. Air-compressor engines have often a negative steam lap. The *in-stroke* is that on which the crosshead moves towards the cylinder. The *displacement* of the valve is measured from its central position. The amount by which the steam port is open when the piston is on dead centre is the *lead* of the valve.

The angle by which the eccentric radius stands ahead of the centre line of the crank is the *angular advance*; with outside steam admission this angle lies between 90 and 180 deg. and is usually about 120 deg. With inside steam admission the eccentric radius stands behind the crank, its *angular lag* being between 0 and 90 deg.

TABLE V.—TYPICAL STEAM CONSUMPTION AND EFFICIENCY DATA FOR VERY GOOD STEAM ENGINES.

Engine.		Steam.		Steam consumption lbs. per i.h.p.-hr.	Heat consumption B.Th.U. per i.h.p.-min.	Thermal efficiency per cent.	Thermodynamic efficiency per cent.	Feed water heating possible, up to °C.
No. of cylinders.	Condensing (C.) or non-condensing (N.C.).	Lbs. per sq. in.	Saturated or superheated to °C.					
1	N.C.	150–180	Satd.	22·5–19·0	445–379	9·5–11·0	64·5–71·6	Over 90°
	N.C.	150–180	300–350	16·2–13·4	355–303	11·9–14·0	76·8–81·0	
	C.	120–150	Satd.	16·8–15·6	335–312	12·7–13·5	52·0–53·4	30 to 45° according to vacuum
	C.	150–180	300–350	11·6–10·0	254–227	16·6–18·6	63·6–67·4	
2	C.	120–180	Satd.	16·8–12·3	335–247	12·7–17·2	52·0–66·5	30 to 40° or by taking steam from the receiver, 60 to 100°
	C.	120–180	270	13·4–10·7	287–227	14·7–18·4	59·1–69·5	
	C.	120–180	300–350	11·2–9·4	245–214	17·3–19·9	68·2–72·2	
3	C.	180–225	Satd.	13·4–11·4	267–227	15·8–18·5	60·6–68·0	20 to 30°
	C.	180–225	270	11·2–10·0	240–214	17·7–19·7	66·7–71·7	
	C.	180–225	300–350	10·0–8·9	220–200	19·2–20·9	71·4–73·5	

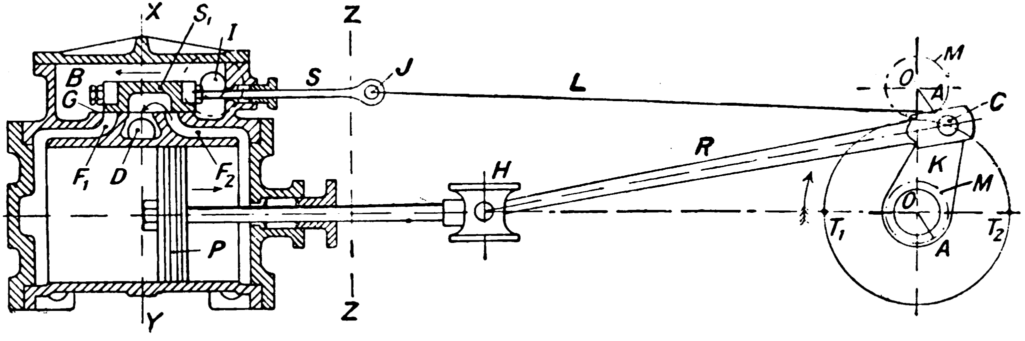

FIG. 23.—ILLUSTRATING SIMPLE SLIDE VALVE.

SIMPLE SLIDE VALVE.

Fig. 23* shows diagrammatically a single-cylinder, double-acting, horizontal engine. The slide valve and steam chest are really at the side of the cylinder; in other words, the diagram to the left of ZZ is a plan view, the remainder being a side elevation with the valve rod L and crank OA (really the radius of the valve eccentric) displaced upwards to prevent confusion. The steam inlet is at I, the ports to the two ends of the cylinder are F_1, F_2, and the exhaust port is D. If a facsimile of the slide valve S be cut from a sheet of paper and moved to and fro along the face G, the passages open to steam flow in various positions can easily be traced. The action is described clearly in detail in Ahrons' book mentioned in footnote.*

In long cylinders the clearance volume may be reduced by using a *divided slide valve.* The two portions of the divided valve are mounted on the same rod, and the steam ports may be kept nearly straight and quite short; the action of the valve is identical with that of the simple valve.

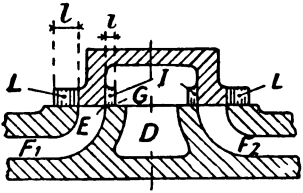

FIG. 24.—SLIDE VALVE WITH OUTSIDE AND INSIDE LAP.

A larger view of a simple "D" slide-valve is shown in Fig. 24, placed centrally with regard to the ports. The outside lap L varies the moment of cut-off and thus determines the extent to which the steam is worked expansively; and in order that admission may occur slightly before the end of the stroke the valve eccentric must be advanced from the position OA_1 (Fig. 25) which it would occupy at right angles to the crank OT_1 if the valve had no outside lap. The advanced position OA_2 gives a projection l = outside lap, and the position OA_3 gives a *lead* e equal to the amount by which the port is uncovered when the piston is at the end of its stroke on its dead

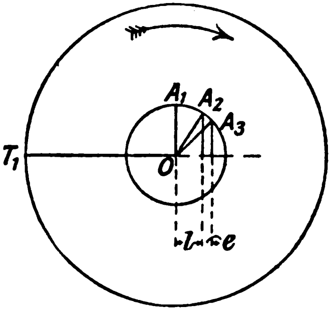

FIG. 25.—ILLUSTRATING ADVANCE OF VALVE ECCENTRIC.

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* This figure and Figs. 24–33, 36, 37, 40, 42–44 are reproduced from *Steam Engine Valves and Valve Gears,* by E. L. Ahrons. (Pitman's Technical Primer Series, 2s. 6d. net.)

centre. The total angle A_1OA_3 is the *angle of advance* of the valve eccentric. The inside lap *i*, Fig. 24, determines the moment at which release occurs and hence varies the extent to which the steam is expanded. The greater the inside lap the later release and the earlier compression occurs. Sometimes the inside lap is nil, and sometimes it is negative, i.e. the valve is cut back to the left of E, Fig. 24, leaving an opening to exhaust when the valve is central; such an opening is called *inside clearance* or *negative inside lap.*

"TRICK" SLIDE VALVE

This valve is illustrated in Fig. 26. Its function is to give more quickly a larger port opening than can be obtained with an ordinary slide valve. A passage P is provided through the body of the valve and steam enters F_1, simultaneously at A and at B through P. The effective port opening is the sum of the openings at A and B and the actual lead is twice the lead at A. The inside lap should close the exhaust side of the port before the edge G opens the port to P, otherwise steam from P passes straight to exhaust.

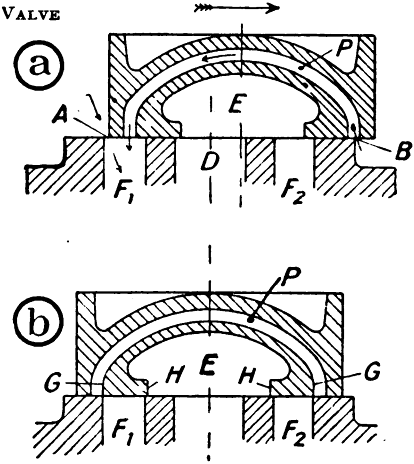

FIG. 26.—TRICK SLIDE VALVE.

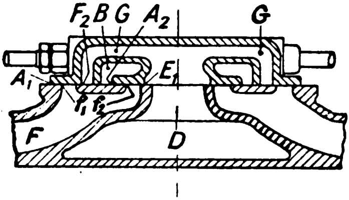

FIG. 27.—DOUBLE-PORT SLIDE VALVE.

DOUBLE-PORT SLIDE VALVE.

Referring to Fig. 27, each steam passage F is provided with two ports. At the instant when the outside lap A_1 uncovers the port f_1, the outside lap A_2 uncovers f_2 and admits steam from the tunnel B through the body of the valve. The exhaust passage G is branched as shown so that exhaust also can take place through both ports $f_1 f_2$. The advantage of this construction is that it gives a large port opening for a short travel. The triple-port valve is on the same principle, (see Fig. 33).

BALANCED SLIDE VALVES.

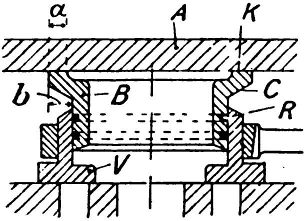

FIG. 28.—SLIDE VALVE WITH BALANCE RING.

A relieving ring or plate is used to exclude live steam from the greater part of the back of the slide valve. A net pressure should be exerted on the valve by the steam in the steam chest in order that the valve may be held firmly on its seating during normal operation. It should be possible, however, for the valve to lift slightly before any dangerous pressure is established by compression of water in the cylinder. The balance ring in Fig. 28 is virtually a floating piston held up against A by steam pressure on C (the width b being greater than a). The piston rings R prevent leakage between B and V. In the Richardson balanced slide valve, Fig. 29, two packing strips D and two similar longitudinal ones fit in grooves in the back of the valve, and are forced against the plate B by springs G. To prevent steam leakage into the space enclosed by the strips D from destroying the "relieving" effect, the hole H is provided so that leakage steam passes to exhaust. The area enclosed by the strips D may = a × transverse breadth of valve.

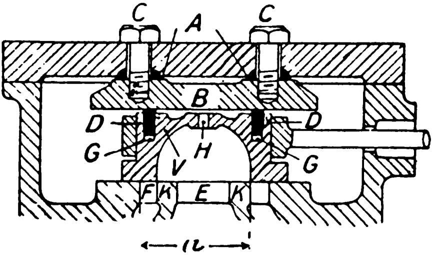

FIG. 29.—RICHARDSON BALANCED SLIDE VALVE.

PISTON VALVES.

The valve rod carries pistons which control steam flow by covering and uncovering ports in the walls of the cylinder within which they move to and fro. These ports FF_1 (Fig. 30) lead to the two ends of the engine cylinder. With *outside admission* live steam is admitted from the outer sides of the valve pistons and exhaust takes place via the space between them. With *inside admission*, as in Fig. 30, live steam admitted at A and, when the valve moves to the left, enters port F as at SS. When the valve moves to the right, exhaust occurs from F to E. The *steam lap* (corresponding to the outside lap of a slide valve) is *l*, and the *exhaust lap* is *i*. The piston valve is perfectly balanced as regards steam pressure, and therefore needs little power to drive it. The pistons bear on renewable liners L and are provided with wide split rings to prevent leakage of steam. In order that these rings may ride smoothly over the ports F, the latter are provided with diagonal bridge pieces which prevent the split rings from entering the ports. A very large port area is obtainable in this type of valve for small valve travel. The advantages of inside compared with outside admission for piston valves is that the former does not expose the valve rod packing to high pressure, high temperature steam.

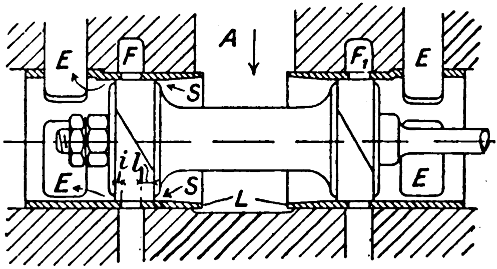

FIG. 30.—PISTON VALVE WITH INSIDE ADMISSION.

VARIABLE EXPANSION VALVES.

In these valves the cut-off can be varied without simultaneously altering the release and compression (as is necessarily the case where simple slide valves are concerned).

MEYER DOUBLE SLIDE VALVE.

Referring to Fig. 31, the lower valve V is a slide valve with steam lap *l* and inside lap *i*. The release and compression are determined by *i*; and the steam lap *l* determines the time at which S_1 is opened to the passage A_1. The upper valve P_1 P_2, operated by a separate valve rod X_1, determines the time at which A_1 is opened to and cut off from the steam chest. The distance *d*, when the expansion valve is central, is the negative outside lap of the expansion valve. The rod X_1 is operated through a swivel-stirrup which permits the rod to be rotated (by a handwheel on the end X_2); when this is done the right

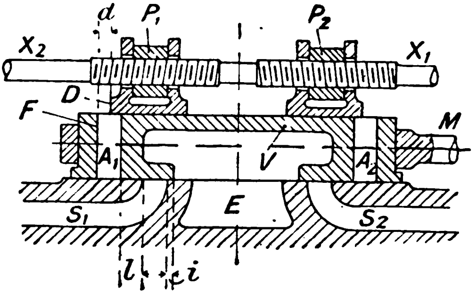

Fig. 31.—Meyer Variable Expansion Valve.

and left-hand threads move P_1 P_2 closer together or further apart, thus producing later or earlier cut-off as the case may be.* The expansion eccentric is usually 170° to 180° ahead of the crank position. The working of the valve is explained in detail in *Steam Engine Valves and Valve Gears*.†

Corliss Valves.

Four independent valves are used to control respectively the admission to and exhaust from each end of the cylinder. Each valve is essentially a cylindrical slide valve (single or double ported) which opens the port when partially rotated by a rod attached to a "wrist plate," the latter being oscillated by an eccentric rod. The exhaust valves are closed by the return motion of the wrist plate rods which open them but the admission valves are generally released from the wrist plate rods by a trip gear or closed by a spring, the moment of tripping being determined by the governor. The advantages of Corliss valves are: (1) The independent setting which can be given to each valve, each being a single-purpose mechanism which does not affect the timing of any other operation in the cycle. (2) The quick action of the valves. (3) The small clearance space. (4) The reduced valve friction compared with ordinary slide valves.

* An alternative method of varying the cut-off is by varying the travel of the expansion valve. Reducing the travel of the latter produces earlier cut-off, whilst increasing the travel retards the cut-off. Hartnell's expansion gear effects this adjustment automatically, the governor being used to move one end of the expansion-valve rod to or from the pivot of a slotted link which is oscillated by the expansion-valve eccentric.

† By E. L. Ahrons, Pitman's Technical Primer Series, 2s. 6d. net.

The four valves for a double-acting cylinder may be actuated by a single wrist plate and eccentric, or two eccentrics may be used (one for admission and one for exhaust) to operate the valves through two wrist plates or equivalent mechanism. The advantage of using two eccentrics is that trip gear can be used on the admission valves with cut-off up to about 75 per cent whereas with a single eccentric the cut-off cannot be later than, say, 45 per cent if the trip gear is to remain effective.

Fig. 32 shows the admission valve mechanism of Messrs. Hick, Hargreaves & Co.'s "crab-claw" Corliss trip gear. Separate eccentrics and wrist plates are used for the admission and exhaust valves. To open the valve, the rod C is moved downwards. The plate D is thus moved counter-clockwise and, with it, the pin E which carries the crab-claw F. The compression spring H keeps the hardened strip J_1 in engagement with J_2, hence G is carried round with D and the valve spindle K is turned, opening the valve. Meanwhile, the extension L of G draws up the rod O and compresses a spring in the dashpot

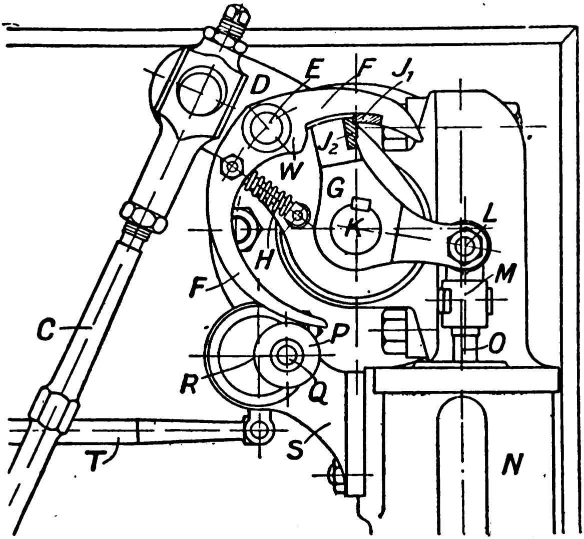

FIG. 32.—ADMISSION VALVE MECHANISM OF HICK HARGREAVES' RADIAL CRAB-CLAW CORLISS VALVE GEAR.

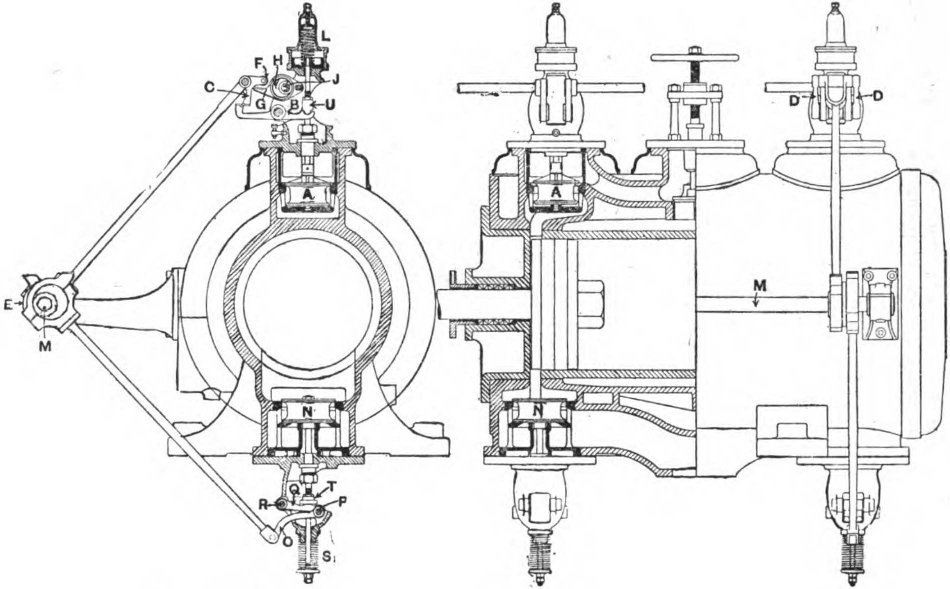

FIG. 34.—MARSHALL DOUBLE-BEAT DROP VALVES AND TRIP GEAR.
(Marshall Sons & Co., Ltd., Gainsborough.)

until the inner arm F comes into contact with G; further movement then disengages C from B. The valve is then closed by the helical spring L, air cushioning being provided. The pressure of F on G has little or no tendency to rotate shaft J, hence the trip gear does not re-act directly on the governor. The exhaust valve N is also operated by an eccentric on the lay shaft through the lever O pivoted at P, the lever Q pivoted at R, and the pad T on the valve stem. The lever Q rests on circular bosses on O when the valve is closed, and the spring S keeps T in contact with the upper side of Q. The upper surface of O is curved so as to give rapid opening and smooth closing of the valve.

The *Paxman-Lentz Positive Valve Gear* (Fig. 35) as used on horizontal engines of the same make (p. 488) is characterized by silent operation due to the elimination of jar or hammering. The valve spindle is ground to fit a long bush and grooves are turned on the spindle to form a labyrinth packing. No soft packing is employed. As shown clearly in Fig. 35, a roller on the valve spindle bears on a cam plate which is actuated by an eccentric on the lay-shaft. Throughout the period when the valve is open the roller is kept firmly in contact with the cam by a spring at the head of the valve spindle. The profile of the cam is such that the double-beat valve is lowered rapidly but without shock onto its seat. Smooth closing is obtained without the use of a dashpot, and there is no hammering action at any speed because the cam does not move out of contact with the

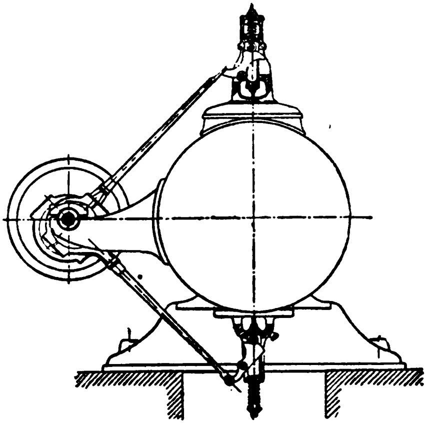

FIG. 35.—PAXMAN-LENTZ VALVE GEAR.

roller until the valve is seated. It is claimed that this system renders permissible the use of piston speeds and revolutions higher than can be attained where trip valve gear is employed. Lentz valves have been substituted for other valves on a number of existing engines with great resultant saving in steam. Each admission valve is free to lift in the event of excessive pressure being developed in the cylinder, and thus acts also as a relief valve. The two admission and two exhaust valves on each cylinder are operated by independent eccentrics. The admission eccentrics are mounted on a sliding block keyed to the lay shaft, and their eccentricity (and therefore the timing of cut off) is controlled by the inertia governor on the same shaft. The tension of the governor spring and therefore the steady speed of the engine can be altered whilst the engine is running.

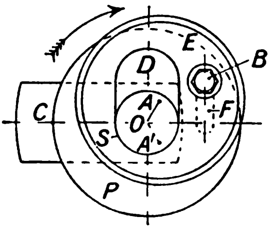

FIG. 36.—SIMPLE ECCENTRIC REVERSING GEAR.

SIMPLE ECCENTRIC REVERSING GEAR.

Fig. 36 shows a simple means of reversal which can be actuated only when the engine is stationary; it is useful for small portable engines. The disc P is mounted on the crankshaft S. The eccentric sheave E is slotted at D and the disc P is slotted at F; E can be clamped to P by the bolt B. In the position shown the eccentric radius is OA. If E be moved downwards so that the other end of D rests on S, the eccentric radius OA^1 and the engine is reversed.

STEPHENSON LINK MOTION.

This mechanism can be used as an expansion gear as well as for reversal. Fig. 37 shows an open-rod link motion as used in

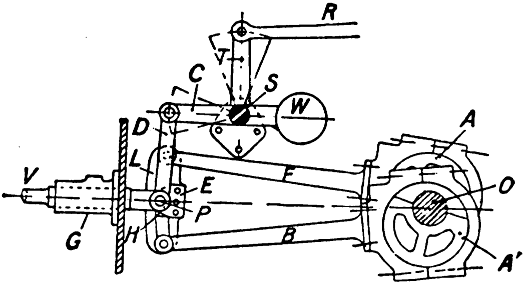

FIG. 37.—STEPHENSON LINK MOTION.

locomotive practice. The rod R turns the shaft S in one direction or the other, and thus raises or lowers the links D from which the link L is hung by trunnions P on the bracket E. The weight of the link motion and eccentric rods is balanced by W. When the link is lowered the fore-gear eccentric rod drives the valve rod V; when the link is raised the back-gear rod is operative. As shown, the mechanism is in mid-gear. The cut-off can be varied between about 17 per cent and 80 per cent by placing the link in intermediate positions. With open eccentric rods the outside and inside leads increase as the cut-off is made earlier (by "notching up"), whereas with crossed rods the leads then decrease. With open rods the steam ports open slightly when the motion is in mid-gear (the maximum opening being then the lead), but with crossed rods the steam ports are not opened by the mid-gear setting. For use with valves having *inside* admission, the eccentric position must be $(270^\circ + a)$ ahead of the crank instead of $(90^\circ + a)$ as for outside-admission valves.

If the link drives the valve rod through a rocking shaft, instead of directly, the necessary eccentric position is $(270^\circ + a)$ ahead of the crank with outside-admission valves, and $(90^\circ + a)$ with inside admission.

In order to determine accurately the distribution effected by this valve gear, the crank circle must be divided into a large number of equal parts, the corresponding positions of the centre line of the link being drawn (by means of a template). The intersection between this centre line and the line of stroke gives the valve displacement which is projected onto a harmonic diagram. By repeating the process for a number of settings of the link the action of the gear can be determined completely.

MacFarlane Gray's construction gives results accurate within 3 or 4 per cent. Referring to Fig. 38, the centre lines of the gear are drawn for the setting under investigation. The points ab are joined by an arc with centre O and radius $sl/2k$. The point p is found such that $ap : pb = AP : PB$. Then the

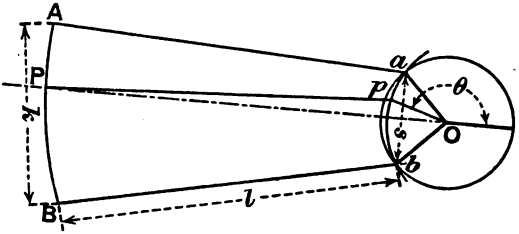

FIG. 38.—DETERMINATION OF EQUIVALENT ECCENTRIC FOR LINK MOTION.

distribution effected is nearly the same as that produced by a simple eccentric of radius Op and angular advance θ.

WALSCHAERT VALVE GEAR.

This mechanism, which is illustrated diagrammatically in Fig. 39, lends itself to good mechanical construction. The eccentric E is at right angles to the crank C and its motion is transmitted to the valve through an oscillating link L (centrally pivoted) and a radius rod FG. The valve rod is connected to one end of the lever ED, which is connected to the crosshead extension AB by the link BD. When the engine is working the lever DE oscillates about F, this motion producing the required lap and lead, and at the same time F is moved to and fro and the valve is given travel varying in magnitude and phase with the position of the block G in link L. The radius rod FG is raised or lowered as required by the link KH and the bell crank on the reversing rod R. When G is at the centre of L the mechanism is in "mid-gear" and the eccentric imparts no travel to the valve. When the crank is on dead-centre, the combination lever DE opens the valve by an amount equal to the lead, this being constant for all positions of the radius rod. The principle disadvantage of this gear is the number of links employed and the unfavourable effect of wear in the various joints. A great advantage of the gear for locomotives is that the links are all outside the frame ; this gear is much used on the Continent and in the U.S.A., and is becoming more popular in this country.

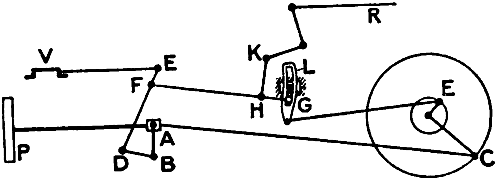

FIG. 39.—DIAGRAMMATIC REPRESENTATION OF WALSCHAERT VALVE GEAR.

JOY'S RADIAL VALVE GEAR.

No eccentric is employed. An elliptical motion is imparted to one end of a link which drives the valve rod. The amplitude of this elliptical motion in the direction of the valve travel equals the length of valve travel and is varied by varying the inclination of the elliptical path. Fig. 40 illustrates the gear as applied to a locomotive. The link C swings about a fixed point O, and the link B is pivoted to C and to a point A in the connecting rod. The link E between N (which drives the valve rod) and B is pivoted to the latter at D. The fulcrum F of the link E is on a slide block in the quadrant L. During running, D moves on an elliptic path of the same vertical amplitude as the path traversed by A. The end of E coupled to N also moves on an elliptic path the inclination of which—and therefore the component in the direction of valve travel—can be changed by altering the inclination of the quadrant L. In the position shown, the crank is on dead-centre and F is co-axial with the spindle about which L is turned, hence the lead is unaffected by the position of L.

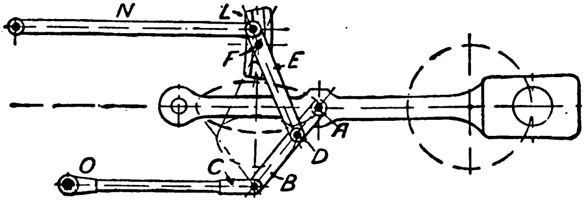

FIG. 40.—JOY RADIAL VALVE GEAR.

Proportions of links in this valve gear: Length of connecting rod $= l = 6\frac{3}{4} \times$ crank radius. Distance from A to crank pin $= \frac{3}{8} l$. AB $= \frac{5}{21} l$. OB $= \frac{1}{2} l$, (approximately); vary to suit position of O. BD $= \frac{5}{8}$ BA. DF $= \frac{5}{18} l$. Perpendicular distance between O and centre line of piston $= \frac{4}{17} l$. FL varies with lap and lead. Length of N variable at will. (*Note.*—B is here taken to be the centre of the pin at the lower end of the link B, Fig. 40.)

VALVE AND PISTON DISPLACEMENTS.

In Fig. 41, the crank is represented by OK and the eccentric radius by OV. A curve of piston displacement may be drawn as follows. The crank circle is divided into, say, 12 equal parts, and the division points 1, 2, 3 are projected to *a*, *b*, *c* by circular arcs of radius equal to the length of the connecting rod (to the scale adopted). A vertical line 0′–12′ is divided into twelve equal parts horizontal line being drawn through the points of division. The points obtained on the horizontal diameter of the crank circle are projected vertically onto the appropriate horizontal line in the lower diagram. Thus the crank position O gives the dead-centre point D.C., *a* gives *a′*, *b* gives *b′*, and so on. Through these points is drawn the curve of piston displacement. The vertical scale of the lower diagram is calibrated to read the crank angle, and the horizontal scale is calibrated in percentage of stroke.

The valve-displacement curve is drawn in the same way; this curve starts from the point v at zero crank angle.

If horizontal scales be added to the lower diagram calibrated in inches of valve and piston travel, the valve displacement x and piston displacement X corresponding to any crank angle θ can be read off at once.

CALCULATION OF PISTON AND VALVE DISPLACEMENT.

If the crank displacement from dead centre be θ deg., and if R = crank radius, and L = length of connecting rod, the piston displacement X is given by—

$$X = R \cos \theta + \frac{R^2 \sin^2 \theta}{2L}$$

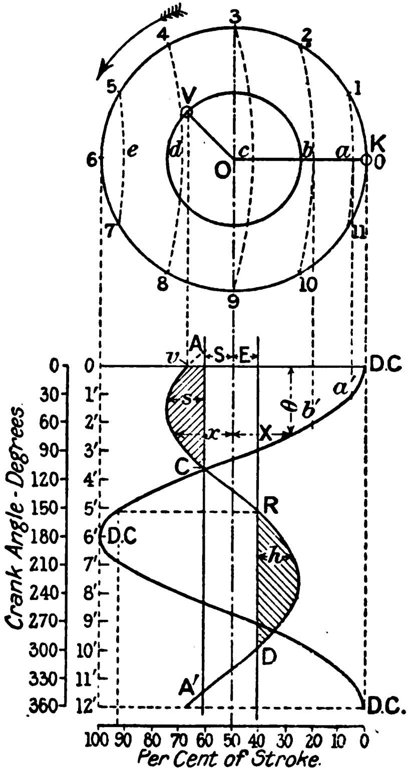

FIG. 41.—HARMONIC VALVE DIAGRAM.

Usually R/L is from $\frac{1}{4}$ to $\frac{1}{6}$ for a main crank and connecting rod. If R/L is very small (e.g. $\frac{1}{20}$ to $\frac{1}{40}$, as for eccentrics and eccentric rods), the second term in the above equation may be neglected. Thus if the angular advance of the eccentric be φ, it is usually sufficiently accurate to take the valve displacement x as—

$$x = r \cos (\varphi + \theta)$$

where r = eccentricity of the valve eccentric.

By equating x to the steam lap, the angle θ can be calculated at which cut-off occurs. From this value of θ the piston displacement X can be calculated. The percentage of stroke at which other events occur in the cycle can be calculated in the same way but it is much easier, and sufficiently accurate, to solve these problems graphically by aid of valve diagrams.

THE HARMONIC VALVE DIAGRAM.

Referring again to Fig. 41, if vertical lines be drawn in the lower diagram at distances from the centre line S = steam lap and E = exhaust lap respectively (to the scales adopted), then admission occurs at A (where the steam lap line intersects the curve of valve displacement), cut-off at C, release at R, and compression at D. The valve opening to steam at any moment is given by the intercept s, and the opening to exhaust by the intercept h. The piston displacement, corresponding to any valve position, is obtained by projecting horizontally from the valve to the piston curve and then vertically to the stroke scale. For instance, in the case illustrated (*not* an actual case) release occurs at 93 per cent of the stroke.

The harmonic diagram is applicable to any geometrical valve gear and should be used whenever special accuracy is desired.

THE VALVE ELLIPSE.

The valve ellipse is the curve obtained by plotting valve displacement against piston displacement (without reference to crank angle). It may be obtained by projection from the harmonic valve diagram and is used considerably in practice.

ZEUNER VALVE DIAGRAM.

This diagram as applied to a simple slide valve is shown in Fig. 42.* For given cut-off, compression and lead, the angle of advance of the eccentric, the outside and inside laps, and the maximum port openings may be found as follows. Taking XX_1 to represent the piston stroke X_1C_1 and X_1D_1 are set out showing to scale the points at which cut-off and compression occur. Arcs C_1D, D_1D struck with radius equal to the length of connecting rod (to scale) give the crank positions C and D at cut-off and compression. Working now with the scale to which XX_1 represents the valve travel, the circle Xa is drawn with radius equal to the lead of the valve. CA is drawn tangent to this circle at a. Through the centre c of AC, draw the line POE. The angle POY = a the *angle of advance* of the eccentric. The radius of the circle *bcd* (drawn tangent to *c* from centre O) represents to scale the *outside lap*. Draw a circle on OP as diameter, then cP represents the *maximum opening of the steam port*. The corresponding intercept on any other radius through

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* Fig. 42 is drawn for the out-stroke of a cylinder located at the left of the diagram.

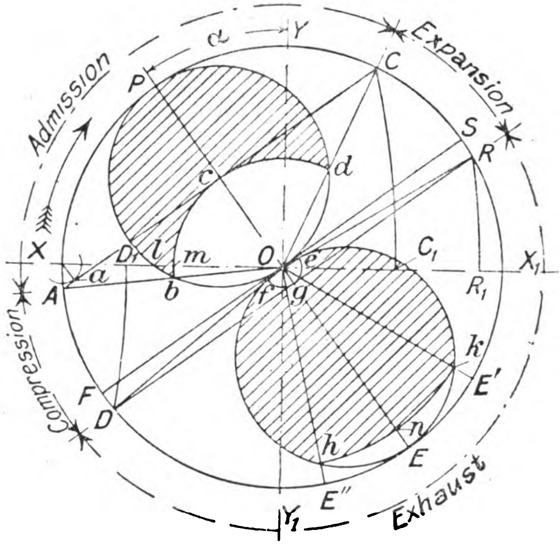

FIG. 42.—ZEUNER VALVE DIAGRAM.

O represents the steam port opening corresponding to that angular position of the crank; e.g. *ml* represents the lead, the crank being on dead-centre X.

From D draw DR perpendicular to OE which it cuts at *g*. O*g* is the *inside lap*. Draw a circle on OE as diameter. Set out *gn* equal to the width of port and, with centre O, draw the arc *hnk*. If *n* falls beyond E, *g* E is the *maximum port opening to exhaust*; otherwise the port is full open to exhaust from *k* to *h*.

REULEAUX VALVE DIAGRAM.

Referring to Fig. 43, S_1 S_2 represents the piston and valve travel (to appropriate scales). P is the piston position at cut-off; draw the arc through B to the line S_1 S_2 of radius = length of connecting rod ; B is crank position at cut-off. Draw the circle S_1 H with radius equal to the lead ; draw BH tangent to this circle ; draw EF parallel to AB, and QR perpendicular to EF. Set out OU = inside lap, and draw DC parallel to EF. The crank position is A at admission ; B at cut-off ; C at release ; D at compression. The angle of advance of the eccentric is a. The steam lap is L. The steam port opening for any crank position K is KM. If UW = width of exhaust port, the latter is full open from Y to Z.

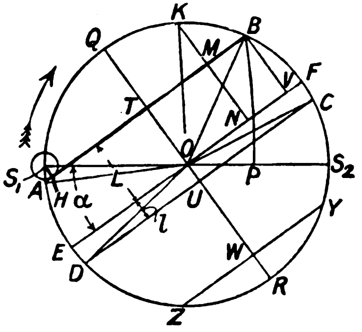

FIG. 43.—REULEAUX VALVE DIAGRAM.

BILGRAM VALVE DIAGRAM.

This is very useful in the design of a new valve, being the only diagram which can be used directly when the valve travel is not known. In all other diagrams the valve travel must be known (or assumed), and a process of trial and error then adopted. *Known data*—Port opening, lead, cut-off, release or compression. *Required*—Valve travel, steam and exhaust laps, angle of advance. In Fig. 44, S_1 S_2 represents the piston travel. OC_1 is the crank position at cut-off, back end. AB is parallel to OS_2 at a distance $e_1 =$ lead at back edge. Preferably the valve scale adopted should be full size. With centre O and radius = back port opening draw the arc *ab*. Bisect C_1AB by DA on which find the centre M of a circle touching *ab*, OC_1, AB. Draw EF through M and O. Then MN = steam lap at back edge of valve, and OM = half the valve travel. $EOS_2 = a$, the angle of advance.

The front lead e_2 is set off, and JL drawn parallel to S_1 S_2. The eccentric circle (radius OM) cuts EF at K. From K as centre draw a circle touching JL. The tangent OC_2 gives the crank position C_2 at cut-off, front end. KP = steam lap, front edge of valve. OP = front port opening. If OQ be the crank position for release, back end, the radius MU of an arc tangent to this line = positive exhaust lap at back edge of valve. An arc drawn with this radius from K as centre gives OR the crank position for compression, back end. Similarly, if X be the release, front end, the radius of the arc tangent to OX from K as centre = positive exhaust lap, front edge of valve ; and the arc drawn with this radius from M as centre gives OW, the crank position for compression, front end. (In the figure, the exhaust laps are taken equal at both ends.)

If the valve has negative exhaust lap the construction is as above except that release, back end, now occurs at, say, OW,

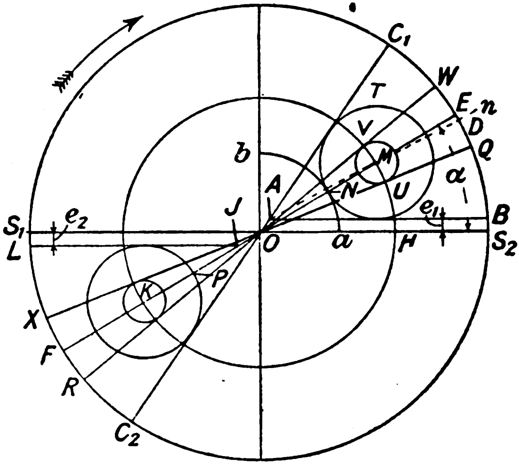

FIG. 44.—BILGRAM VALVE DIAGRAM.

and the compression later than before at OX (the arc with radius MV now being drawn above OF).

GOVERNING ENGINE OUTPUT.

The i.h.p. of an engine at constant speed may be varied by changing the area of its indicator card: (*a*) By varying the initial pressure by throttling, the cut-off being constant. (*b*) By varying the cut-off, the initial pressure being constant.

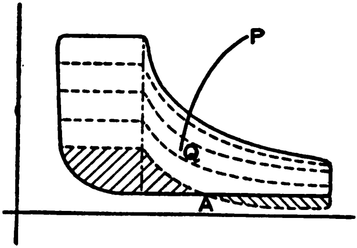

FIG. 45.—GOVERNING BY THROTTLE, WITH CONSTANT CUT-OFF.

Control by Throttle. Referring to Fig. 45, the full-power card is shown by a solid line and the reduced-power cards are dotted. With low-admission pressure the expansion line may fall below the back pressure line at

A, i.e. beyond A the work done is negative. The net work done per stroke is then proportional to the difference between the shaded areas. Knocking is liable to occur at A due to the reversal of stress. Almost any throttle-governed engine has a negative loop in its card when running light, the only work done being that required to overcome friction. If the engine is much too large for its work, a negative loop appears in the card at relatively high loads. Though the volume of steam admitted per stroke is constant, the dryness fraction is higher at lower admission pressures. The equivalent volume of d.s. steam is represented by P at full-power and by Q under light-load conditions, i.e. the volume of d.s. steam per stroke decreases with the horse-power. Throttle governing with constant cut-off proves very economical in Willans' high-speed engines. Within limits the economy is higher the earlier the constant cut-off.

The total weight of cylinder feed per hr. plotted against the i.h.p. gives nearly a straight line which is known as the *Willans' Line* (Fig. 46). The weight represented by the intercept *a* is the light load steam consumption.

Control by Cut-off. Fig. 47 illustrates this method of governing, the admission pressure being practically equal to boiler pressure throughout. The point X representing the volume of dry steam moves inwards for some time as the cut-off is made earlier, i.e. the steam efficiency of the engine is improved. Beyond a certain point (say 3) the steam becomes wetter again as the cut-off is made still earlier; the point X then moves outwards from the position of maximum economy. Cut-off at 70 per cent or later may be used temporarily when starting heavy loads. Maximum efficiency in steam locos is generally obtained at about 20 per cent cut-off.

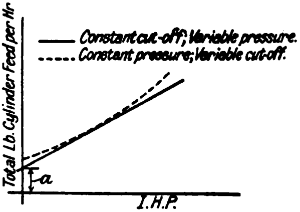

FIG. 46.—WILLANS' LINES.

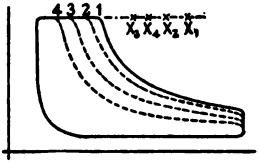

FIG. 47.—GOVERNING BY CUT-OFF WITH CONSTANT ADMISSION PRESSURE.

TYPES OF ENGINES.

Reciprocating steam engines may be classified according as they are: (1) Single or double acting, i.e. steam acts on one or both sides of the piston. (2) Vertical or horizontal line of stroke. (3) Low or high speed, as judged by crankshaft r.p.m.

In a low-speed engine the r.p.m. is of the order of 100 whilst in a high-speed engine it is from 250 r.p.m. to 500 r.p.m. or higher. If the stroke is very long the piston speed may be higher in a "low-speed" engine than in a high-speed, short-stroke engine. (4) Simple or compound, the latter term here embracing triple and quadruple expansion. (5) Non-condensing or condensing.

The term "superheated engine" often used in commercial literature means simply that superheated steam is employed; the engine may be of any one of the above types. Engines using steam at about 150 lbs. or higher pressure are often called high-pressure engines but this again is no real basis of classification. With the notable exception of locomotive engines practically all compound engines are operated condensing.

HORIZONTAL ENGINES.

The horizontal engine is admirably adapted to stationary installations in mills, factories, power stations, etc., where efficiency and overall economy are important and the output

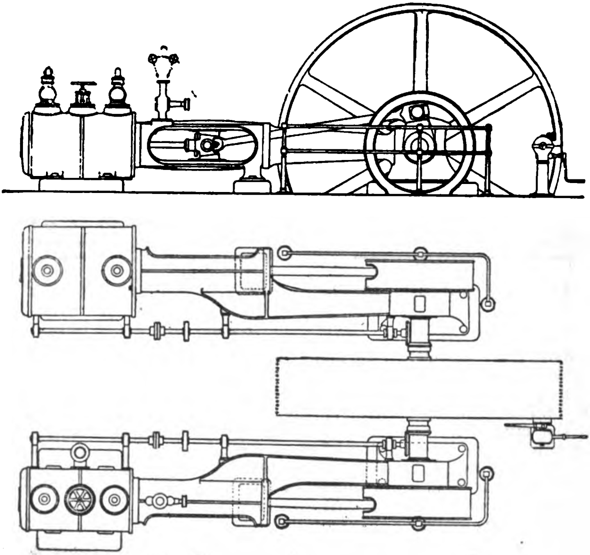

FIG. 48.—GENERAL ARRANGEMENT OF CROSS-COMPOUND HORIZONTAL ENGINE.
(Marshall, Sons & Co., Ltd., Gainsborough.)

required does not justify the installation of steam turbines. Engines of this type are usually low-speed units. They occupy more floor space than a vertical engine for the same speed and much more floor space than a high-speed vertical engine. On the other hand, they are easily accessible at all parts and they are exceptionally reliable and durable. High-speed vertical engines have an advantage where floor space is very valuable or where a high speed of rotation is required for direct driving of dynamos, and vertical engines are more convenient than the horizontal type for marine purposes, but for general industrial service the low-speed horizontal engine enjoys well-deserved popularity. Engines of this type are usually (1) single cylinder, condensing, or non-condensing engines; or (2) compound, condensing or non-condensing engines. Wherever possible condensers should be used and, unless the load is very variable and cheap fuel is available, compound engine should be used in preference to a simple engine.

In small single-cylinder horizontal engines for outputs up to about 100 h.p., steam pressures from 80 to 120 lbs. per sq. in., and speeds from 120 to 220 r.p.m., a bent crankshaft may be carried between bearings in a continuous bedplate extending the whole length of the engine, or a short shaft for the flywheel and crank-pin disc may be carried by a bearing in a girder extension of the crosshead slide.

A typical cross-compound horizontal engine is shown diagrammatically in Fig. 48.* This engine and simple and tandem compound engines of the same type are standard in the following sizes.

Cylinder diameter and stroke.	R.p.m.	B.H.P., using steam at 100 lbs.	
		Condensing.	Non-condensing
		Normal/Max.	Normal/Max.
7″ × 20″	135	14/31	14/27
to 22″ × 48″	75	180/410	180/355
7″ and 13″ × 20″	135	42/58	41/45
to 22″ and 38″ × 48″	75	485/670	462/509

Using steam at 80 lbs., the smallest simple engines are suitable for 10–15 h.p., and using steam at 150 lbs. the largest compound engines are suitable for 500–1,000 h.p. Coupled tandem compound engines are used extensively for loads up to about 2,000 h.p. The engines are suitable for boiler pressures up to 150 lbs. per sq. in. and for piston speeds up to 600 ft. per min.

* In these notes no attempt is made to describe the products of every engine builder in this country. Where specific makes are mentioned this is only by way of introducing typical data. Reference must be made to text-books or to the technical press for full treatments of all makes.

The Marshall engine illustrated has double-beat drop valves on both the steam and the exhaust sides, each being separately adjusted (see also p. 472). Disengagement of the tripping levers does not directly react on the governor. A pendulum-type governor is chain driven from the eccentric shaft and maintains speed constant within 4 per cent from no-load up to a cut-off at ⅝ stroke. A mechanical oil pump (driven by eccentric and ratchet) is used to lubricate the cylinders, and oiling rings are used in the main bearings. Drip lubricators are provided for all the other moving parts. The engine can be reversed in a few minutes if required.

From 12 to 20 per cent saving in coal is effected by condensing, compared with non-condensing. A simple jet condenser may be installed immediately adjoining the engine, the air pump being driven by a tail rod from the main piston. The quantity of water required is from 28 to 30 times the weight of steam passing through the engine: the jet condenser is capable of raising its water from a depth of 16 to 18 ft. (straight lift). Where space is valuable or where the condensing water is drawn from a low level, the condenser may be placed below the engine, the air pumps being driven by L-levers from the tail rod. This arrangement has the advantage that the piston speed of the air pump can be reduced.

The intermediate receiver in compound engines is jacketed with boiler steam and a by-pass is provided so that steam at boiler pressure can be supplied to the l.p. cylinder for warming and starting.

A vertical tubular type feed heater may be connected to the engine exhaust to recover a proportion of heat from the steam on its way to condenser or atmosphere. Feed water at 180° F. may thus be obtained.

The Paxman-Lentz horizontal engine is built as a simple engine and as cross-compound and tandem-compound engines in standard sizes which cover the following ranges—

Cylinder diameter and stroke.	R.p.m.	I.h.p. using steam at 100 lbs.	
		Condensing.	Non-condensing
		Normal/Max.	Normal/Max.
13″ × 24″	190	110/180	114/173
to 29″ × 54″	90	593/967	608/936
9″ and 15½″ × 18″ (tandem)	210	120/132	105/115
or 13″ and 21½″ × 24″ (coupled)	190	277/310	245/265
to 29″ and 48″ × 54″ (tandem or coupled)	90	1470/1650	1300/1420

Different powers are, of course, obtained if the engines be run at other than standard speeds. Using steam at 170 lbs. instead of 100 lbs. per sq. in., the horse-power of the compound engines is increased roughly 50 per cent when non-condensing and about 25 per cent when condensing. The mechanical efficiency of a particular 600 h.p. cross-compound condensing engine of this type was 92 per cent at 1¼ times full load; 91½ per cent at full-load; and 86 per cent at ½ load. The maximum powers listed under "condensing" in the above table are the maximum at which a reasonable economy may be obtained; the absolute maximum of the condensing engine is slightly higher than that of the non-condensing engine. It will be seen that this engine is relatively a high-speed, high-power type. It is designed to give best results with high-pressure, superheated steam. Patent metallic packing and forced lubrication for the valve chests, packing and pistons are employed. Other special features are the use of a patent inertia governor and positively operated double-beat valves (see p. 475). Steam consumption data for this engine operating *condensing* are given in Table IV (p. 463) and range from about 10½ lbs. per i.h.p.-hr. with 150° F. superheat to 13 lbs. in small engines using saturated steam. With steam at 170 lbs. per sq. in. and 150° F. superheat the steam consumption per i.h.p.-hr. *non-condensing* is about 14 lbs. in a 200 h.p. unit and 13½ lbs. in a 400 h.p. engine.

MARINE ENGINES.

Recent improvements in geared turbines will probably result in turbine drive gradually becoming standard on all but the smallest cargo vessels. At present, however, vertical triple-expansion reciprocating engines are used on the majority of cargo vessels. These engines have either three or four cranks (the four-crank type being easier to balance) and run at low speed (60 to 80 r.p.m.) to suit direct coupling to large propellers. The cylinders are side by side parallel to the axis of the vessel and are carried by massive columns bolted to a heavy bedplate which also carries the crankshaft bearings. The surface condenser is mounted along the side of the engine at about half the total height of the latter. The use of superheated steam is becoming common. Piston valves are commonly used on the h.p. and i.p. cylinders and a balanced slide valve on the l.p. cylinder. Typical dimensions, etc., are as in Table VI. (See page 490.)

SMALL VERTICAL ENGINES: OPEN TYPE.

These are low- and medium-speed machines for small workshops, etc., requiring from 5 to 15 h.p. Generally non-condensing, using steam at 80–120 lbs. per sq. in., and running at 150–220 r.p.m. In some designs there is only a single crosshead guide, in which case the engine may not be reversed; a "keeper" guide surface prevents the crosshead from lifting should the thrust be reversed temporarily.

TABLE VI. TYPICAL DATA FOR ENGINES OF CARGO VESSELS.

I.h.p.	Cylinder diameters.			Stroke.	Deadweight tonnage.	Speed of vessel
	H.p.	I.p.	L.p.			
	Ins.	Ins.	Ins.	Ins.	Tons.	Knots.
400–600	11–13	18–22	30–35	20–27	600	9½
650–750	15–17	25–29	40–49	27–33	1000–2000	9½
1400–1800	21–25	33–37	56–68	36–45	3000–5000	9½–11
2200–2700	25–27	41–44	68–74	45–51	5000–10000	11–12
3000–3500	27–30	45–48	75–79	52–54	8000–10000	12
5000 quadruple-	23	32 & 45	65	48	11,000	13

VERTICAL MEDIUM-SPEED ENGINES.

The enclosed compound Corliss engine illustrated in Fig. 49 is typical of this class, and is available for outputs from 300 to 600 i.h.p. The particular engine shown has cylinders 17 ins. and 30 ins. diameter × 24 ins. stroke and runs at 150 r.p.m. Numbers of these machines are used to drive electric generators and for mill driving. In the latter application, the engine is sometimes direct coupled to the main lineshaft. The smaller floor space occupied is an advantage of the vertical engine compared with horizontal engines of similar speed and output; also, there is no tendency for the weight of the piston to wear the cylinder oval as in horizontal engines. Two oscillating valveless oil pumps provide forced lubrication at 10–15 lbs. per sq. in. in the crosshead pins, crank pins and main bearing. Positive-feed oil pumps lubricate the cylinders. The Corliss valves are actuated by the crab-claw gear described on p. 471. The eccentrics operating the valve gear are keyed to a layshaft which is chain-driven from the crankshaft.

HIGH-SPEED VERTICAL ENGINES, OPEN TYPE.

Robey engines of this type, used extensively for ship lighting and wireless installations, are available in standard sizes from 3″ × 3″ to 23″ × 14″ single cylinder for pressures from 40 to 150 lbs. per sq. in., yielding from 2 to 140 b.h.p. The smallest engines run at 500–750 r.p.m. and the largest at 150–225 r.p.m. Compound engines of the same type are built with cylinders from 4″ and 7″ × 4″ up to 14½″ and 23″ × 14″, for pressures from 80 to 150 lbs. per sq. in., and outputs from 4 to 220 b.h.p. at speeds from 200–550 r.p.m. in the smallest to 150–225 r.p.m. in the largest sizes. A crankshaft throttle governor adjustable whilst the engine is running is simple and reliable; expansion governing may be used in the larger engines.

VERTICAL ENCLOSED ENGINES.

This type is specially suitable for use in confined spaces and where cleanliness is important; common applications are to driving ship and other electric lighting sets, pumps, fans,

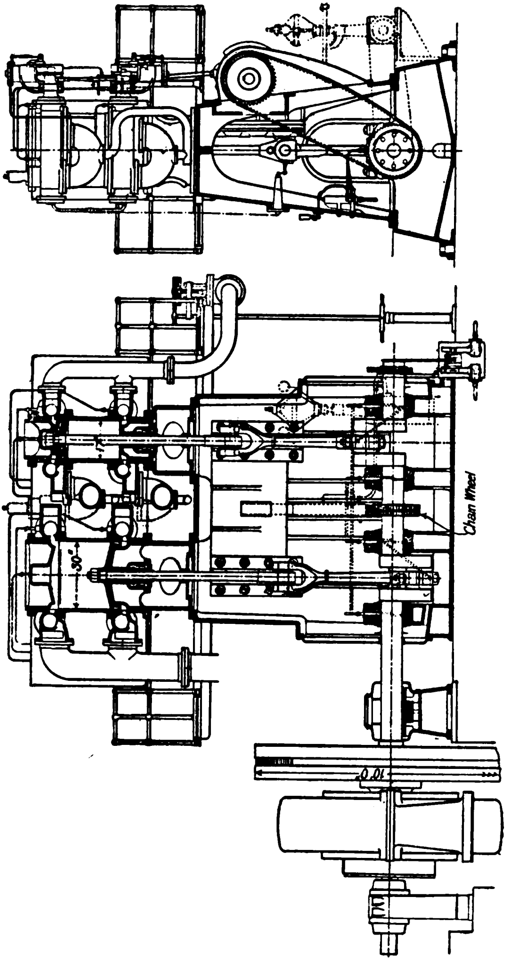

FIG. 49.—VERTICAL ENCLOSED COMPOUND CORLISS ENGINE.
(*Hick Hargreaves & Co., Ltd., Bolton.*)

factories, mills, etc. The larger engines of this type are very suitable for electric power stations, when the output required does not justify the use of steam turbines.

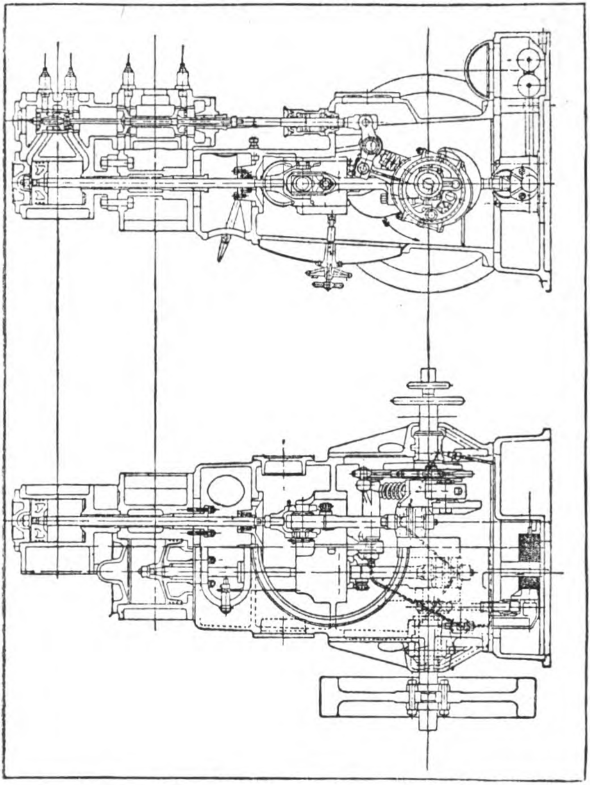

FIG. 50.—SISSON HIGH-SPEED VERTICAL ENCLOSED ENGINE. (*W. Sisson & Co., Ltd., Gloucester.*)

Sisson Engines. Outputs available range from 7 to 400 b.h.p. The engines are self-lubricating on the splash and forced systems, and are fitted with automatic expansion and compression crankshaft governor. Typical steam consumptions are given in Table VII, in which: (1) SC denotes single-cylinder, single-crank, double-acting engine. (2) TSC denotes tandem-compound, single-crank, double-acting engine. The steam economy effected by this engine as compared with the SC engine is obvious from the table. (3) DC denotes compound, double-crank, double-acting engine. The main features of this engine are shown in Fig. 50. The engine is of the opposite crank type with partially superposed h.p. cylinder and is specially suitable for superheated steam since the h.p. piston rod passes through a long gunmetal-lined trunk and the steam heated portion of the rod does not come into the packing space. Piston valves are used and there is only one valve spindle stuffing box and that is exposed only to receiver steam.

To prevent misapprehension it should be noted that Messrs. W. Sisson & Co., Ltd., also make open-type compound and triple expansion engines for factory driving, etc., and marine engines up to 1,200 i.h.p. on single screw or 2,400 i.h.p. on twin screw.

Belliss and Morcom Engines. These are vertical, enclosed, high-speed, double-acting engines various types of which are made for non-compound, compound, or triple expansion working. In each type a special feature of the construction is the automatic forced-lubrication system as originated by this firm in 1890. A special oil pump, without valves or packings, is driven from the slide valve eccentric and, drawing oil through a gauze strainer from an oil well in the crank chamber, the pump delivers oil under a pressure of from 10 to 30 lbs. per sq. in. to the main bearing caps. Thence oil flows to the crank-pins, crosshead pins and guides, governor joint pins, and valve motion as shown in Fig. 51. This illustration shows in section the *C-type compound engine.* The slide valves for both cylinders are operated by a single eccentric rod, the l.p. valve being placed above the h.p. valve and working in the same valve chamber. The h.p. valve takes steam on the inside and exhausts directly to the l.p. valve which cuts off on the outside edge. There is only one exhaust outlet and no outside receiver pipe. The cranks are set at 180 degrees, and steam is admitted simultaneously to the top of one cylinder and the bottom of the other, thus contributing towards balancing the reciprocating parts and reducing the load on the bearings. The centrifugal crankshaft governor keeps the speed variation within 3 per cent between full-load and no-load and the set speed can be adjusted whilst the engine is running. Metallic packing is used on the h.p. piston rod and valve rod and soft packing on the l.p. rod. The distance between cylinders and casing stuffing boxes exceeds the length of stroke hence the piston rods cannot carry oil from the crankcase into the cylinders. The pendant crosshead guides are self-centring.

The *V-type compound engine* is similar to the C-type engine except that the high and low pressure slide valves are in separate

TABLE VII.—TYPICAL TEST DATA FOR VERTICAL ENCLOSED ENGINES USING SATURATED STEAM

(*W. Sisson and Co., Ltd., Gloucester.*)

Type of engine.		Full load, b.h.p.	Saturated steam at lb./sq. in.	R.p.m.	Lb. steam per b.h.p.-hr. at fractions of full-load.					Vacuum; Ins. of mercury.
					¼	½	¾	1	1¼	
SC–4″	Non-cond.	16	140	660	56·0	49·0	42·2	39·4	37·2	—
SC–6″	Non-cond.	40	80	500	57·5	44·7	41·6	39·0	37·6	—
	Condensing	40	80	500	40·0	34·2	31·5	31·7	32·2	25
TSC–5″	Non-cond.	30	150	510	43·5	34·2	30·8	30·0	28·2	—
	Condensing	30	150	510	33·1	28·8	26·2	24·6	24·2	26
DC–6″	Condensing	64	110	605	31·4	23·1	20·7	19·8	20·3	27
DC–7″	Non-cond.	118	168	558	35·7	25·9	23·6	22·75	22·8	—
DC–8″	Non-cond.	135	152	507	33·2	25·9	24·1	23·7	—	—

valve chambers, the valves being set between the cylinders in a plane at right angles to the crankshaft and at an angle on either side of the vertical. The valves are operated radially, from the shaft as centre, by separate eccentrics. The h.p. valve cuts off on the inner edges and exhausts directly to the l.p. valve. Variable expansion gear, hand or governor operated, is fitted to the h.p. valve and overloads of 25 per cent can thus be carried.

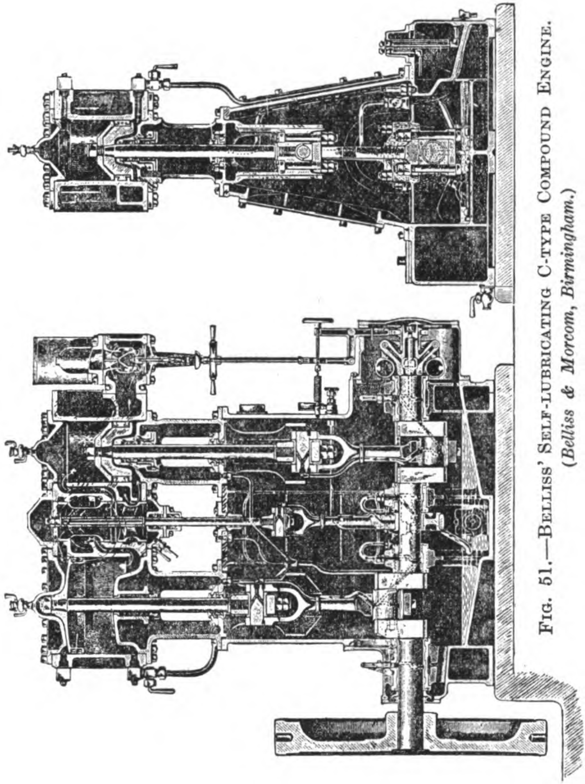

FIG. 51.—BELLISS' SELF-LUBRICATING C-TYPE COMPOUND ENGINE. (*Belliss & Morcom, Birmingham.*)

The ***T-type triple expansion engine*** is broadly similar to the

C-type compound engine but has, of course, three cylinders driving cranks at 120 deg. Approximate steam consumption data for this type of engine are given in Table VIII.

TABLE VIII.—TYPICAL TEST DATA FOR VERTICAL ENCLOSED TRIPLE EXPANSION ENGINES USING SUPERHEATED STEAM

(*Belliss and Morcom, Ltd., Birmingham.*)

Full load, b.h.p.	Steam.		R.p.m	Lbs. steam per b.h.p.-hr. at fractions of full load.				Vacuum; Ins. of mercury
	Lbs. per sq. in.	Super-heat ° F.		½	¾	1	1¼	
300	160	307	475	—	—	10·75	—	26·7
715	186	100	250	12·8	12·4	12·3	—	27·6
1050	165	155	250	11·8	11·4	11·5	11·8	25·5
865	169	117	230	—	—	11.95	—	26·6
1075	180	226	230	11·0	11·0	10·9	—	26·0

Simple, compound, and triple expansion Belliss engines are used extensively with Belliss air compressors, the air cylinders being placed above the steam cylinders and the air pistons being carried by extensions of the steam piston rods. These compressors are fitted with Rogler-Hoerbiger plate valves and are built for pressures up to about 100 lbs. for supplying compressed air tools, etc., and for pressures up to 3,000 lbs. per sq. in. for torpedo service, etc.

Output, Weight, etc. Non-compound Belliss engines for pressures about 80–90 lbs., non-condensing are available in standard sizes from 32 b.h.p., 650 r.p.m. to 720 b.h.p., 300 r.p.m. Compound C-type engines, operating condensing or non-condensing with steam at about 150 lbs., are available in standard sizes from 32 b.h.p., 650 r.p.m. up to 250 b.h.p., 428 r.p.m.; the V-type engine is standard up to 570 b.h.p., 375 r.p.m. Three-crank compound and triple expansion engines are available up to 2,000 b.h.p. or over. Roughly, the compound engine averages 70 lbs. weight per b.h.p. or, including a dynamo and base plate, 112 lbs. per b.h.p.

OVERTYPE AND UNDERTYPE ENGINES.

In the overtype equipment, the engine is mounted on top of the boiler which supplies it with steam. This arrangement is employed in many tractors and steam vehicles and is useful in such cases and in small portable and stationary plants. The boiler has to be designed to carry the weight of the engine, and there must be an additional margin of safety to allow for stresses set up by the engine when working; for obvious reasons every effort is made to reduce to a minimum unbalanced stresses from this source. In the *undertype* equipment the boiler is of locomotive type and the engine is placed under the barrel of the boiler (as in railway locos). This arrangement is compact and is suitable for stationary plants of moderate capacity (up to

500 h.p.). The boiler has not to carry the engine and is not subjected to any stresses from the latter; the weight of the boiler helps to provide a steady foundation for the engine.

An overtype or undertype steam engine set may be regarded as a very satisfactory alternative to a gas or oil engine; the overload capacity of the steam engine is much higher than that of internal combustion engines, and it is generally considered that steam plant is more easy to manipulate and maintain. The use of superheated steam makes possible a coal consumption of about 1¼–1½ lb. per b.h.p.-hr. condensing, or 1¾ lb. per b.h.p.-hr. non-condensing. Miscellaneous waste fuel can often be used to advantage.

The Garrett overtype engine is made: (1) Single cylinder for outputs from 21 to 62 b.h.p. (25–70 b.h.p. emergency), 210–200 r.p.m. (2) Compound, non-condensing, 55 to 390 b.h.p. (65 to 470 b.h.p. emergency), 200–160 r.p.m. (3) Compound, condensing, 65 to 520 b.h.p. (75 to 600 b.h.p. emergency), 200–160 r.p.m. The sectional illustration (Fig. 52) shows clearly all the essential features of the construction. The entire furnace, fire tubes, and smoke-box tube plate can be withdrawn for cleaning purposes. A jet condenser is employed. The distribution valve gear is of the balanced piston valve type, the crankshaft governor regulating the h.p. cut off. The superheater chamber may be arranged for direct uptake to a 60 ft. steel chimney or for side- or down-take to flues discharging into a 90 ft. chimney.

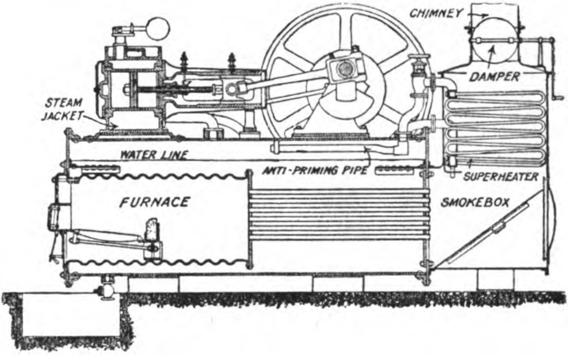

FIG. 52.—DIAGRAMMATIC SECTION THROUGH "GARRETT" ENGINE AND BOILER.
(*Richard Garrett & Sons, Ltd., Leiston.*)

Robey compound overtype engines are available in standard sizes from 40 to 450 b.h.p. normal (58 to 570 b.h.p. emergency) output, jet condensing; and from 30 to 335 b.h.p. normal (45 to 420 b.h.p. emergency), non-condensing. Feed water is heated to 170° F. by heat abstracted from the exhaust on its way to the condenser. The jet condenser and air pump maintain a vacuum sufficient to draw injection water from a depth of 15 ft. (straight lift); about 40 galls. of water is required per b.h.p.-hr. If preferred, a surface condenser can be used; about 70 galls. of circulating water is then required per b.h.p.-hr. The boiler is of the fire tube type, with corrugated circular fire-box and a superheater in the smoke-box. An extension fire-box is fitted if low-grade fuel is to be burned. The engine is of the compound horizontal type with side stay rods between cylinders and crankshaft bearings. The cylinders are bolted to pressed steel seatings riveted on the boiler, and the crankshaft bearings are bolted to a casting which in turn is bolted to steel brackets riveted on the boiler. Belt drive from the flywheel is usual up to 250 h.p.; rope drive for higher powers. A balanced piston valve is used on the h.p. cylinder and a slide valve on the l.p. cylinder. The crankshaft governor varies the h.p. cut-off and the expansion plate on the l.p. valve gear enables the cut-off to be varied from $\frac{1}{8}$ to $\frac{3}{4}$ stroke.

The Marshall "locomobile" is a semi-portable power plant comprising a fire-tube boiler with superheater, a compound over-type engine and (where required) a condenser—usually of the jet type. Standard sizes are available for normal outputs from 50 to 250 b.h.p. (with 25 per cent continuous and 40 per cent momentary overload capacity) at speeds from 215 to 165 r.p.m.; 180 lbs. per sq. in boiler pressure. The compactness and strength of construction are obvious from Fig. 53. Exceptionally low fuel consumption is required per b.h.p.-hr. Tests on a No. 8 locomobile developing 307 b.h.p. condensing and 241 b.h.p. non-condensing show the mechanical efficiency to be about 95 per cent and the heat consumption (*at the grate*) about 15,990 B.Th.U. per b.h.p.-hr. condensing and 19,550 B.Th.U. per b.h.p.-hr. non-condensing, corresponding to 1·15 lbs. and 1·4 lbs. of coal respectively at 14,000 B.Th.U. per lb. During these tests the engine was supplied at 180 lbs. per sq.in. 155·6° F. superheat (condensing), 172° F. superheat (non-condensing); vacuum 27 ins., condensing; r.p.m. 165; feed temperature 121° F. (condensing), 188° F. (non-condensing).

As shown in Fig. 53, the engine is built as a separate unit, and is carried on a strong steel superstructure attached to the boiler. The crankshaft carriages and crosshead guides are exceptionally heavy, so that no bending stress is imposed on the boiler by the engine forces. The support for the crank carriages allows the boiler to expand without affecting the engine centres or cylinder clearances. The boiler flue and tube unit and the superheater are easily removed for cleaning, etc., and an extension fire-box is used for low-grade fuels. Flue gases can be by-passed past the superheater when raising steam. Balanced piston valves are used on both cylinders. A centrifugal crankshaft governor controls the h.p. admission, and the l.p. valve is operated from the air pump eccentric (see Fig. 53) with a cut-off suiting the average load on the engine. The location of the jet condenser and air pump is as shown.

The Marshall S-type portable and semi-portable engines are combined with loco type boilers to form overtype units. The single cylinder, non-condensing engines, using saturated steam, are made for from 9 to 39 b.h.p. (13 to 50 b.h.p., max. continuous output). The engine with 8½ ins. cylinder and 12 ins. stroke yields 32·2 i.h.p. when supplied at 150 lbs. per sq. in. The water consumption is 19·25 lbs. per i.h.p.–hr.; 20·17 lbs. per b.h.p.–hr. (test data). Using coal of 14,500 B.Th.U. per lb., the fuel consumption would be 2·06 lb. per b.h.p.-hr. (ash

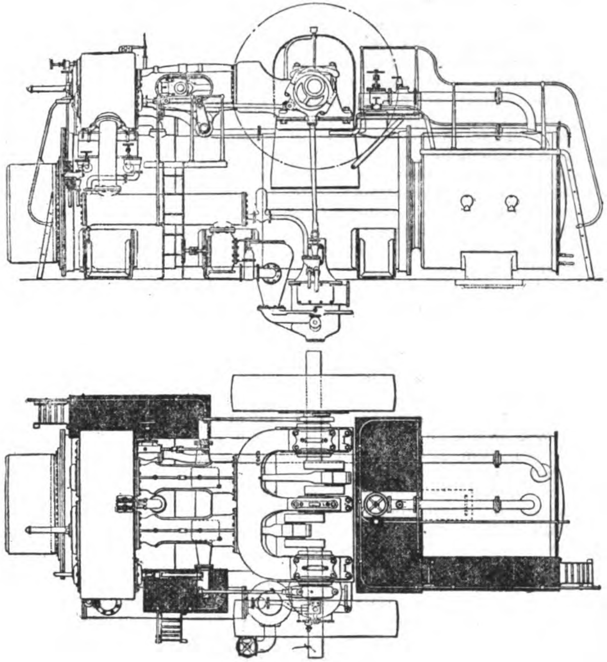

Fig. 53.—Elevation and Plan of Marshall "Locomobile."
(*Marshall, Sons & Co., Ltd., Gainsborough.*)

and clinker being deducted). Balanced piston valves are used, and the mechanical efficiency of the engine is 95·4 per cent.

"UNIFLOW" ENGINE.

In the "uniflow" engine steam is admitted at each end of the cylinder, but leaves via ports at the centre of the cylinder which are uncovered by the piston when it approaches the end of the stroke. In each stroke the steam flows in one direction only, from the end of the cylinder to the centre, hence the name "uniflow." The principal advantage of this is that the ends of the cylinder are never exposed to the relatively cool expanded steam, hence heat loss from steam to cylinder walls is reduced and, with it, the cooling or condensation of the admitted steam. As a consequence, single stage expansion in a uniflow cylinder compares very favourably with expansion in two or more stages in compound engines; the losses between cylinders and in the receivers of multi-cylinder engines are here eliminated. Drop valves or piston valves are used for admission in uniflow engines. If the main piston acts also as exhaust valve it must be nearly equal in length to the length of stroke and the cylinder must be nearly twice as long as the stroke. Due to the length of piston no tail rod is required. The length of the exhaust ports is about 10 per cent of the stroke. The timing of exhaust (about 90 per cent stroke) and compression (about 10 per cent stroke) is fixed once for all, if controlled by the main piston. Fig. 54 shows the distinctive features of an engine of this type, viz., double beat inlet valves, long piston, and central exhaust; live steam jacketing of the inlet end of the cylinder; restriction of wet steam to the centre of the cylinder, and the reduced condensation resulting in close approximation to adiabatic expansion.

If exhaust is controlled by an independent piston valve, the main piston and the cylinder may be of normal length. Single-acting vertical uniflow engines have recently been developed for marine service.

The uniflow principle was patented by the British engineer L. J. Todd, in 1885, and was put into practice by Prof. Stumpf (Charlottenburg) in 1908; since then its manufacture has been taken up by a number of firms. The following particulars are of interest.* The clearance volume in condensing uniflow engines is about 2 per cent of the piston displacement, corresponding to the high compression ratio of 1 : 46. A high vacuum is required to obviate abnormal compression pressures; relief valves are fitted as a safeguard against the latter when starting or in event of loss of vacuum. Exhaust is to a condenser placed immediately below the cylinder, a feed water heater being sometimes interposed. Early cut off is required to obtain the full range of expansion in a single cylinder; at full load, with steam at 170 lbs. per sq. in., cut off is at 8 to

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* See also papers by H. Pilling, Manchester Assocn. of Engineers, March, 1920, and F. B. Perry, Inst. Mechanical Engineers, July, 1920.

10 per cent. Positive valve gear and a sensitive, powerful governor are required. The moving parts are heavy (and therefore of high inertia) to withstand the high initial load on the piston. In non-condensing uniflow engines the clearance space is about 18 per cent of the piston displacement.

The uniflow engine is essentially a condensing engine and is not recommended for reversing drives. It is very suitable for textile, paper, and timber mills ; for non-reversing rolling mills ; and for dynamos. It can be built for slow and variable speeds, but is more suitable for high and fairly constant speeds. The largest uniflow engine built up to 1914 was a mill engine rated at 4,000 h.p., 120 r.p.m., but capable of supplying 6,000 h.p. continuously and 8,000 h.p. in emergency ; the engine had a single cylinder 1,700 mm. (66·9″) diam., 1,400 mm. (55·1″) stroke. A larger uniflow engine, built by Galloway, Ltd. (Manchester) for driving sheet mills has a cylinder 60 ins. diameter × 72 ins. stroke and runs at 28 r.p.m., using steam at 160–180 lbs. per sq. in., 150° F. superheat. The economy due to the uniflow principle can be realized in small as well as in large engines.

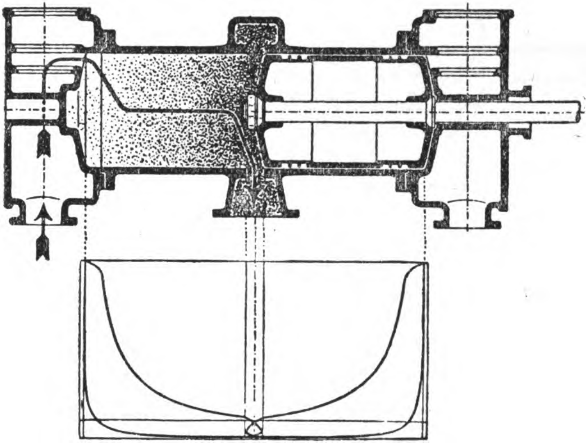

FIG. 54.—SECTIONAL DRAWING OF "UNIFLOW" CYLINDER.

(*Galloway, Ltd., Manchester.*)

The Robey uniflow engine, illustrated in Fig. 55, is covered by the introductory remarks above. As shown by the figure, exhaust and compression are controlled by the main piston,

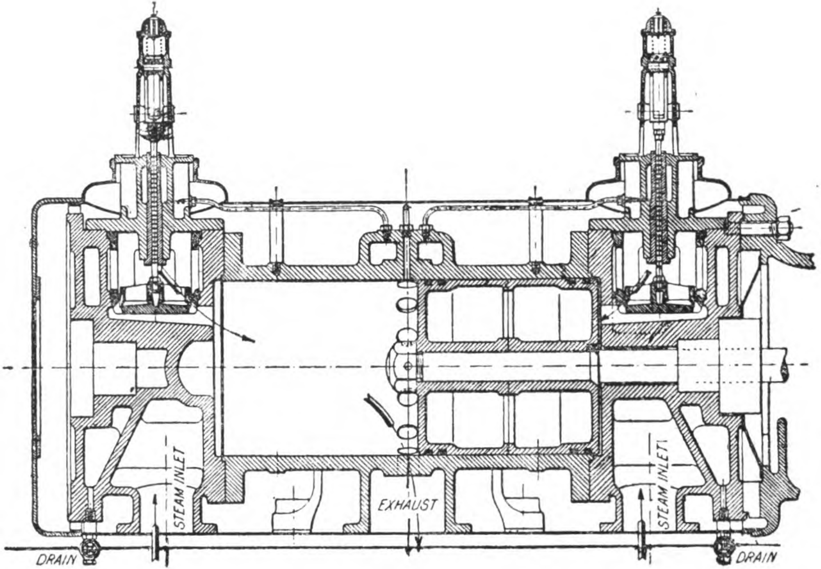

FIG. 55.—ROBEY "UNIFLOW" ENGINE.
(*Robey & Co., Ltd., Lincoln.*)

which is therefore of nearly half the cylinder length. In addition to high-power units, the Robey uniflow engine is also supplied in an undertype combination complete with superheater and jet condenser. Standard sizes of this combination range from 24 h.p. to 160 h.p. normal output (38 to 248 h.p. in emergency) working non-condensing; and from 30 to 200 h.p. normal (48 to 310 h.p. emergency) working condensing.

The Hick-Hargreaves (Tabourin) uniflow engine illustrated in Fig. 56 uses double-beat equilibrium or drop piston admission valves, and the exhaust through the central ports is controlled by an independent piston valve as shown. This permits the main piston to be of normal length and weight, and eliminates the difficulties and risks associated with an abnormally heavy piston in respect of cylinder wear. The exhaust ports being only in the side of the cylinder, it is easier to keep the lower part of the cylinder properly lubricated. The auxiliary valves mounted across the bottom of the cylinder act as relief valves, and can be used to remove dirty oil from the cylinder. They also can be used as auxiliary exhaust valves after the main exhaust has closed, thus permitting the compression to be reduced to 8 or 10 per cent for non-condensing service, and permitting the release to be varied.

The use of a long-lift plate valve instead of a double-seat admission valve offers considerable advantages. As described by Stumpf,* such a valve has a flat seating, but is formed somewhat like a blunt needle-valve on the cylinder side of the plate, so that good stream-flow is obtained when the valve is open. The valve is driven from a shaft running at twice engine speed and has four times the lift otherwise obtainable. A plate valve of 100 mm. diameter can be used instead of a double-beat valve of 200 mm. diameter, and the total clearance volume is between ¾ and 1 per cent of the cylinder volume. The plate valve need not be ground-in at working temperature, and it is tight at all pressures and temperatures. It is claimed that the steam consumption is reduced by from 0·9 to 1·1 lb. per h.p.-hr. compared with uniflow engines with double beat valves.

ENGINES WITH HIGH EXHAUST PRESSURE.

If exhaust steam be used for heating, evaporating, etc., the engine operates with a back pressure of 5 to 12 lbs. per sq. in. above atmosphere. The steam consumption per i.h.p.-hr. is high, but the overall thermal efficiency is good, because the latent heat of the steam can be utilized in heating service, but not in an engine. The following data† (Table IX) relate to the *h.p. cylinder only* of compound engines, in which the conditions are much the same as in a single-cylinder engine operating against high back pressure. Due to the lower friction, etc., in a single-cylinder engine, at least as low steam consumption should be obtained as shown in the Table, say, 25 to 30 lbs. per i.h.p.-hr.

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* *Zeits. Vereines deutsch Ing.*, May 7, 1921, pp. 492–494.

† See also *Mechanical World*, 12th March, 1921, p. 171.

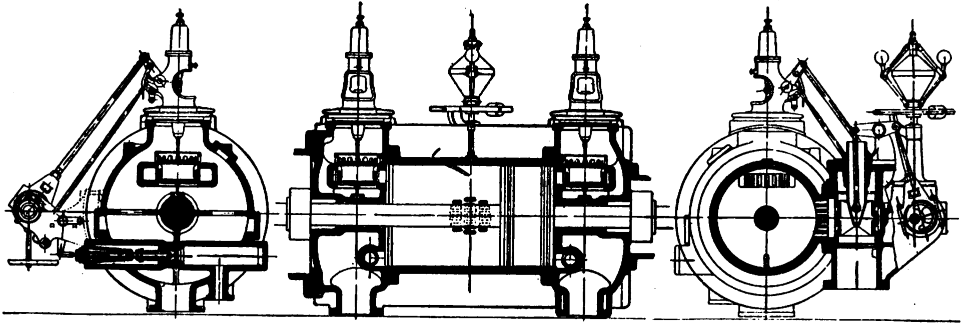


FIG. 56.—LONGITUDINAL AND CROSS SECTIONS OF TABOURIN-TYPE UNIFLOW ENGINE.
(*Hick Hargreaves & Co., Ltd., Bolton.*)

TABLE IX.—STEAM CONSUMPTION OF ENGINES OPERATING WITH HIGH BACK PRESSURE

Type and size of engine.	R.p.m.	Max. and min. pressures from card. Lbs. per sq. in. above atmos.	I.h.p. (h.p. cylinder only).	Dry, satd. steam per i.h.p.-hr. lbs.	Efficiency ratio. Per cent.
Corliss vertical, 14″ and 27″ × 36″ . .	71	120–10	89	26·6	75·7
Corliss tandem, 16″ and 32″ × 42″ . .	75	120–10	210	26·0	77·6
Corliss cross compound, 26″ and 46″ × 60″ . .	52	80–2	228	27·7	73·1
Pair tandem, double slide valves, 30″ and 50″ × 72″	52	82–6	958	28·9	77·6
Tandem, piston valve, 23″ and 43″ × 60″ .	53	80–5	549	31·8	69·9
Cross compound double slide valves, 35″ and 58″ × 60″	45	63–4	432	35·1	70·5

Single-cylinder engines operating with high back pressure ought to be used more extensively in connection with industrial heating service. In its passage through the engine the steam loses only the thermal equivalent of the mechanical power developed. The remainder of the heat is available for the heating service.

[The author takes this opportunity of expressing his indebtedness to Mr. Chas. B. Clapham, B.Sc., for assistance in revising the above notes, and for a number of useful data not commonly available.—R.E.N.]

STEAM TURBINES

THE PRINCIPLE OF THE STEAM TURBINE AND SUB-DIVISION INTO TYPES

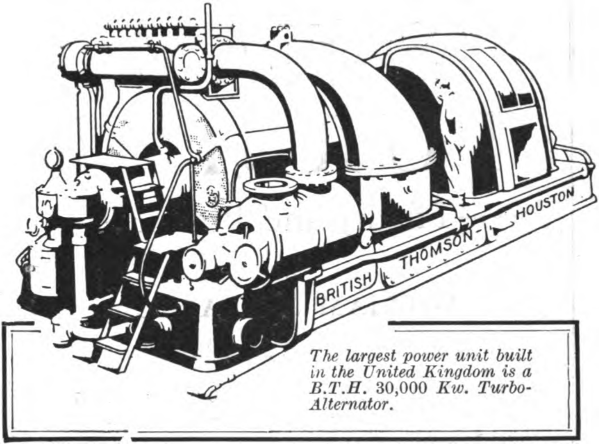
STEAM Turbines are commercially so-called "Impulse" or "Reaction" Turbines, to distinguish two main types, but the names are more "popular" than scientific.

Case (1). "*Simple Impulse*" *Turbine.* The simplest form of steam turbine is a steam "windmill." Suppose that, in Fig. 1, we have a wheel W mounted on a spindle, and carrying projections or buckets B all round the periphery. At A, suppose that we have a nozzle (or nozzles) directing a continuous jet (or jets) of steam along an oblique tangent, on to the buckets. Then, when it meets the entrance edges, the steam will be deflected round the curved surfaces of the buckets, and if the wheel be free to rotate, it will do so in the direction BN. In other words, the curved surfaces of the buckets exert a force on the steam and deflect it out of the straight path ABX, whilst the buckets are themselves deflected backwards by the "reaction" of the steam upon them. The steam jets from the stationary nozzles are said to give an "impulse" to the wheel.

Case (2). *Simple* "*Reaction*" *Turbine.* Suppose now, that instead of placing the nozzles stationary at A, one is placed on every bucket, supplied with steam through the shaft itself, and arranged so as to "fire" in the direction BC in Fig. 2. Then the "kick" of the nozzles will have a component in the direction BN, and the wheel will rotate if the resistance to motion be not too great. It is, in fact, driven by the "reaction" of the steam issuing from the moving nozzles. The steam jets are said to exert a "pure reaction" on the wheel.

Case (3). *Simple* "*Impulse*" *and* "*Pure*" *Reaction combined.* Let us suppose that, in Fig. 3, we have a combination of cases (1) and (2). That is, suppose that stationary nozzles A direct steam in the direction AB on to the buckets, that the jets of steam are deflected along the curved surfaces BD (thereby imparting backward motion to the buckets), and that as they reach the outlet edges D, a further expansion takes place, thus speeding up the steam along the deflected path DC, so that the additional "reaction" on the rotating wheel (as in case (2)) will produce an extra driving force on the buckets. Then we have a simple turbine in which the arrangements in cases (1) and (2) are combined.

It will be seen that the *underlying* principle is the same in every case, that is to say the heat energy in the steam is first converted into kinetic energy in the jets, and mechanical work is then obtained from it by means of a rotating shaft, carrying a wheel and buckets. Further, in both cases (1) and (2), it is really the reaction of the steam which produces the driving torque. In the "impulse" turbine the reaction is due to the



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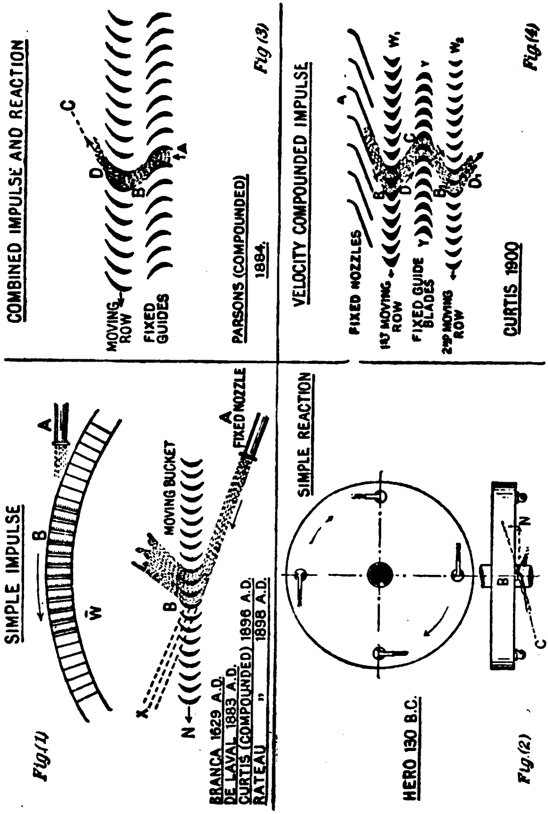
steam jets being slowed down, whereas in the "reaction" turbine, it is due to the steam jets being speeded up through the blade passages.



TURBINE PRINCIPLES ILLUSTRATED BY DIFFERENT TYPES.

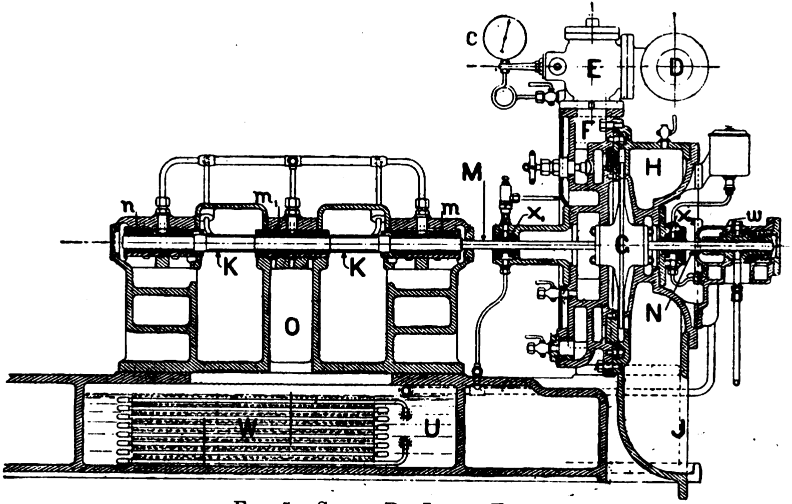
The turbines described in the above three cases are all "simple" turbines, in which the shaft is driven by the torque exerted on one wheel, carrying one row of buckets (or nozzles in the case of the "reaction" turbine).

Type (1) is represented in commercial practice by a turbine such as the De Laval, which is shown in longitudinal section in Fig. 5 (a design of *Messrs. Greenwood & Batley, Ltd., Leeds*) It will be seen that there is only one set of nozzles, followed by only one row of moving blades. The nozzle velocity is that due to the expansion of the steam from nearly the initial boiler pressure right down to the condenser pressure, and the resulting jet-velocity is so enormous that it is not possible to run the wheel fast enough to obtain the best efficiency, which depends on the ratio of the mean blade-speed to the steam-jet velocity. That is to say the ratio

$$\frac{\text{blade speed}}{\text{steam speed}} = \frac{U}{V} \qquad (1)$$

has a certain value corresponding to the most efficient conversion of heat energy into mechanical work. It can easily be shown by mathematics that if the buckets were shaped so that they turned the steam jets round through 180 degrees, the theoretical efficiency would be unity when the above-mentioned "velocity ratio" has a value of 0·5. Actually, the efficiency attainable is much less, and the finite blade angles which have to be adopted in practice (in order to get the steam into and out of the blades) make the best "velocity ratio" rather less than 0·5. Now, the steam velocity arising from an expansion in one stage right down to the exhaust pressure (say 180 lbs. dry saturated down to 28-in. vac.) may amount to as much as 4,000 ft. per sec., whereas the highest permissible mean blade speed is about 1,000 ft. sec., and is only attainable with a wheel of small diameter, running at a very high rate of rotation. If a larger wheel were used, running at a lower rate of rotation, but with the same mean blade-speed, then the stresses set up would be the same as in the case of the small diameter wheel, at similarly situated points, but it would not be possible to make so reliable a forging in the larger size, and consequently the safety factor would have to be increased, so that a still lower mean blade speed would have to be permitted. In addition, excessive surface frictional losses (depending on the 5th power of the dia.) would still further reduce the efficiency. With a 7-in. wheel running at 30,000 r.p.m. the mean blade speed is 920 ft. sec., so that the velocity ratio would be 920/4,000 = 0·23, whereas for the best efficiency, a velocity ratio of about 0·45 is required. Thus we really require a mean blade speed about twice as high.

To get over this difficulty, and to enable more moderate blade speeds to be used, Curtis adopted the principle of "velocity compounding." Suppose that, in Fig. 4, the steam, after leaving the first row of buckets along the deflected path DC, is deflected into the original direction once more by means of a row of fixed buckets YY. It can then be directed on to the buckets of a second wheel W_2, mounted on the same shaft, and will, in thus being deflected again by these buckets, assist in driving the shaft.



D = steam strainer
E = governor valve
F = steam chest
G = turbine wheel
H = exhaust chamber
J = exhaust pipe
K = pinion
M, N = two halves of turbine spindle
O = gear case
W = spiral oil cooler
U = oil well
X_1 X_2 = right- and left-hand-tightening bushes
w = ball bush
c = pressure gauge
$\left.\begin{matrix} m \\ n \end{matrix}\right\}$ = bearings

FIG. 5.—SMALL DE LAVAL TURBINE.
(*Messrs. Greenwood & Batley, Ltd., Leeds.*)

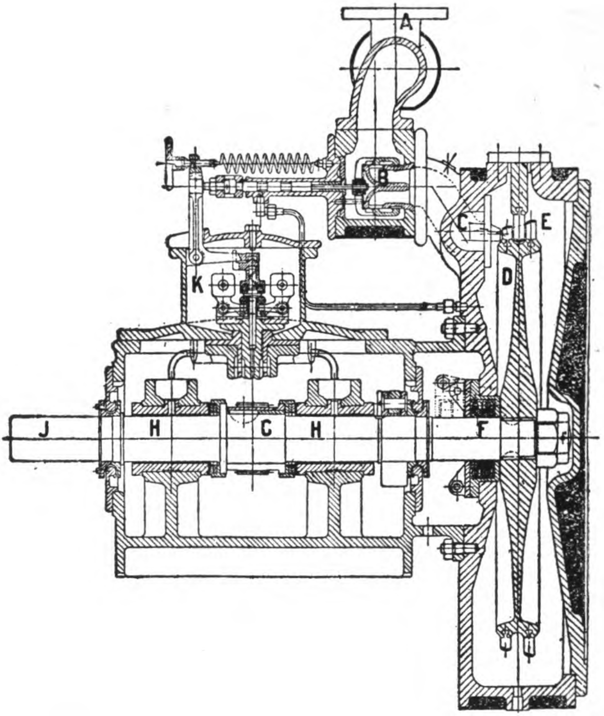


FIG. 6.—BROTHERHOOD SINGLE IMPULSE WHEEL WITH TWO ROWS OF REVOLVING BLADES.

A = Steam inlet pipe.
B = Governor throttle valve.
C = Nozzles.
D = Impulse Wheel.
E = Impulse blading.
F = Carbon segment gland.
G J = Turbine shaft.
H = Bearings.
K = Speed-regulating governor.

In practice, a second set of "guides" may be fitted, and a third row of moving buckets. We may even have a four-row wheel, that is with four rows of moving buckets.

This principle of placing two or more rows of "moving blades" in front of the nozzles, with fixed "guide blades" in between each row, is known as "velocity compounding." It has the advantage of making the turbine very short. In fact, theoretically, five or six rows on a suitable diameter could be made sufficient to deal with the entire heat drop, and at the best velocity ratio; but in practice, it is impossible to obtain good efficiency even at the best velocity ratio. It has been found by long experimenting, that to obtain good efficiency, the jet velocity, and the deviation of the jet by the blades, must be kept moderate, otherwise the losses due to friction, shock and eddying, are serious, more especially if the high velocity and deviation occur simultaneously.

It can readily be shown mathematically that if, in a "velocity compounded" wheel, there are *n* moving rows, then the theoretically best velocity ratio is given by the fraction $\frac{1}{2n}$. With a four-row wheel, then, if the initial jet velocity be 4,000 ft. sec., we shall require a mean blade speed of one-eighth or 500 ft. sec., which is a speed readily attainable in practice, with ordinary materials. Unfortunately, however, the circumstances are very unfavourable to good economy, for we start off with very high jet-velocity, and then subject the steam to seven reversals in succession. The unavoidable friction, shock and eddying, that takes place in the blading makes the efficiency very inferior—of the order of 40 per cent—so that in practice a four-row wheel is very rarely used. In fact, the efficiency obtainable with velocity compounding alone—even with a two-row wheel—is never really satisfactory, so that velocity compounding alone is never resorted to except in cases where steam economy is of secondary importance, and shortness and simplicity are the chief desiderata. This state of affairs often arises with small lighting sets on ships, and where the auxiliaries of a plant are steam driven, because it is possible to utilize the exhaust steam from the small turbine to heat the feed water for the main engines. The small turbine exhausts at a pressure a few lbs. above atmosphere. Fig. 6 is a design due to Messrs. Peter Brotherhood, Ltd., Peterborough.

PRESSURE COMPOUNDED TURBINES.

A pure "pressure compounded" impulse steam turbine is one in which the total pressure drop is divided up over a series of compartments. Each compartment contains one wheel, carrying one row of "blades" only, and all the wheels are mounted on the same shaft. Steam is admitted to the first wheel through nozzles, does work on it, and leaves the first compartment through passages which at one and the same time form the exhaust passages to that compartment, and the inlet passages to the next. Thus expansion goes on from "stage" to "stage" in the successive compartments.

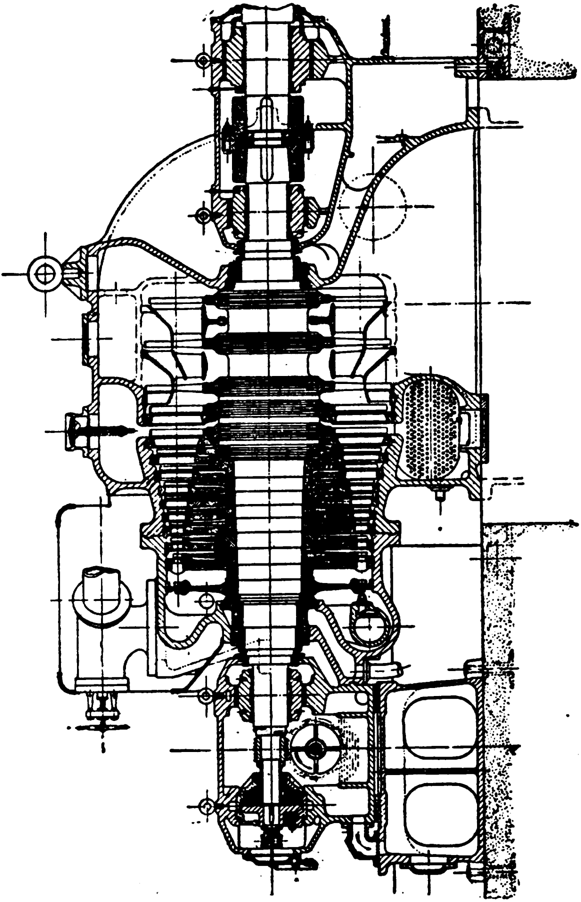


FIG. 7.—METROPOLITAN VICKERS COMPOUND IMPULSE TURBINE.

Each compartment forms a "simple" turbine of the "impulse" type, and, as we have seen before, the best velocity ratio is about 0·45, a value which is easily attained when the heat drop is divided up in this manner.

Fig. 7 shows a modern "Rateau" compound impulse steam turbine, as built by the *Metropolitan Vickers Co., of Manchester*. It will be seen that "pressure compounding" alone is used, and this arrangement gives very good economy.

The expression "*compound steam turbine*" is usually taken to mean a "pressure-compounded" turbine, but the turbine may also be "velocity compounded," either in some or in all of its compartments, although to obtain an economy competitive with the "pure pressure compounded" type, a velocity wheel can only be used in the first compartment. The facts underlying this conclusion are as follows:

The foremost feature of the compound impulse* steam turbine is its comparative shortness, which is a very desirable property, especially when it is a question of getting as powerful a turbine as possible into a given floor space. To effect this result, the wheels are made the same diameter throughout, because fewer wheels are then required to efficiently expand the steam right down to the exhaust pressure, than would be the case if the wheels were made smaller at the inlet end. This is because the higher the mean blade speed, the higher the steam velocity must be in order to run the wheels at a velocity ratio of 0·45. Hence the larger the heat drop over each set of nozzles must be, in order to produce that velocity.

Now the choice of the wheel diameter depends on the output required, and on the vacuum, that is, on the total volume of steam to be passed per second through the blading of the last wheel. We shall return to this point later. Having fixed the diameter of the wheel at the exhaust end, therefore, if we then make all the other wheels the same diameter, then at the inlet end, if we admit the high pressure steam through nozzles all round, the nozzle height required to admit the necessary steam becomes very small, and in many cases quite impracticable. Since, however, there is little or no pressure drop over the moving blades in an impulse turbine, it is possible to admit steam through a small sector only, and thus obtain a reasonable nozzle height at the inlet end. This arrangement is known as "partial admission."

One serious source of loss in impulse turbines, however, is due to the "windage" of the impulse discs, due to the fact that they are revolving in steam.† Partial steam admission also brings with it another serious loss in efficiency, because when steam is admitted through a small sector only, then only those blades actually passing the nozzles at any instant will be "active," and the remainder will have to be driven round in an

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* Originally due to Curtis, both Velocity Compounding and Pressure Compounding.

† *See* list of references at end of article.

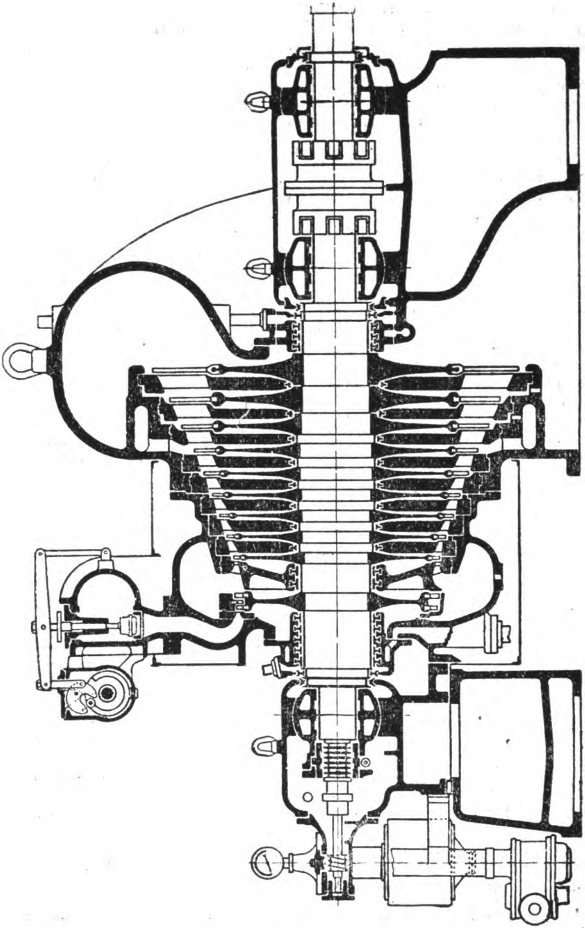


FIG. 8.—BRITISH-THOMSON-HOUSTON COMPOUND IMPULSE TURBINE.

atmosphere of steam. The power absorbed in doing this may be very considerable. Every effort has, therefore, to be made to reduce the practice of "partial admission" to a minimum, and to begin as soon as possible to admit steam all round the wheels. Now the losses mentioned above are directly proportional to the density of the steam in which the wheels revolve. If we replace, say, the first three single-row wheels of a "pure-pressure compounded" turbine, by a two-row velocity wheel of the Curtis type, then the steam can be made to enter the next compartment at the same pressure as before, that is to say the Curtis wheel will be revolving in steam at a much lower density than in the first two replaced compartments. Hence, although the blade efficiency of the Curtis wheel may be considerably less than that of the Rateau wheels, yet the net efficiency may be quite as good, because not so much of the output is absorbed in driving the disc and idle blades round in steam, and because the steam leakage from the first compartment to the next, at the point where the shaft passes through them, and through the external shaft gland, is also reduced owing to the fact that the pressure inside it is now equal to that in the third compartment, when the wheels are all single-row. A turbine of this type in which a Curtis wheel is used at the commencement, is shown in Fig. 8. It is a compound impulse steam turbine whose foremost characteristic is its extreme shortness, and the design shown is due to the *British-Thomson-Houston Co., Ltd., of Rugby.*

PRESSURE COMPOUNDED "REACTION" STEAM TURBINES.

The type described in case (3) is only used in conjunction with pressure compounding. In this design, the compartments become virtually non-existent and are replaced by "pairs" of "reaction blading," consisting of a ring of guide blades followed by a ring of moving blades of equal height. Simple mathematical theory shows, and experiment bears out, that when "simple impulse" and "pure reaction" are combined the best velocity ratio is about unity, that is to say, the velocity of the steam jets impinging on to the moving blades must be about equal to the mean blade speed. When this is the case, the turbine is driven practically by the "reaction" of the jets of steam issuing from the rows of moving "blades." The steam jet velocity, for a given mean-blade speed, is thus lower than in any of the other types, and as we have seen before, this is a circumstance favourable to high inherent blade efficiency. It is also satisfactory from a mechanical point of view, for a high velocity jet of steam tends to erode the surfaces on to which it impinges. On the other hand, not only is a low steam speed required, but low blade speeds also are inevitable in the earlier stages, because in these earlier stages much smaller mean diameters have to be used than are possible (and absolutely necessary) at the exhaust end of the turbine. The turbine is thus inevitably longer than one of the "impulse" types of the same output and speed of rotation, because more "compartments"

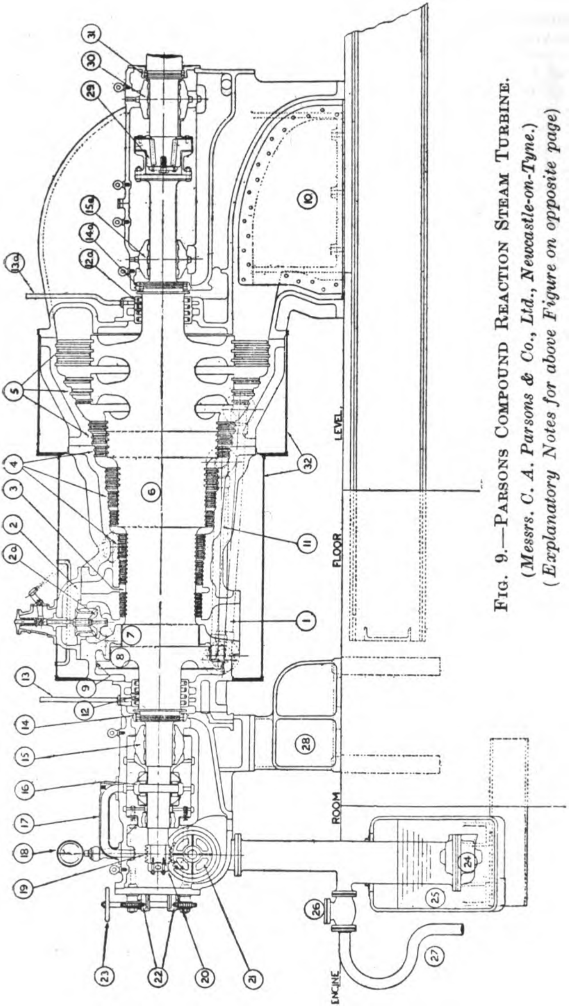


FIG. 9.—PARSONS COMPOUND REACTION STEAM TURBINE.

(Messrs. C. A. Parsons & Co., Ltd., Newcastle-on-Tyne.)

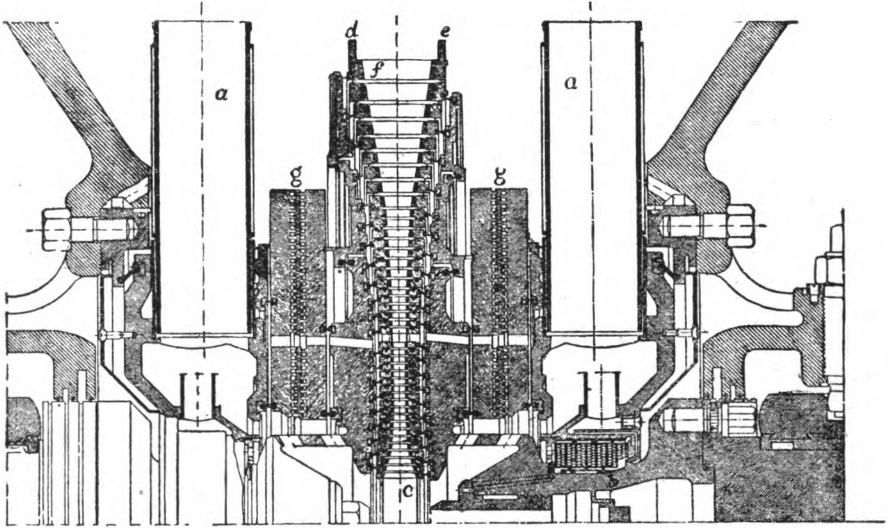
(Explanatory Notes for above Figure on opposite page)

1. Steam inlet from steam chest.
2a. Overload byepass valve.
2. Byepass passage to lower stage of blading.
3. Byepass steam-inlet to blading.
4. " End-tightened " reaction blading.
5. " Ordinary " radial-clearance reaction blading.
6. Turbine shaft.
7. High pressure dummy piston.
8. Intermediate pressure dummy piston.
9. Low-pressure dummy piston.
10. Turbine exhaust.
11. Equalizing pipes from dummy pistons (shown dotted).
12, 12a. Carbon segment glands.
13, 13a. Gland vapour pipes.
14, 14a. Oil baffles.
15, 15a. Turbine main bearings.
16. Pivoted thrust block for adjusting axial working-clearances.
17. Pipe conveying oil discharged from thrust bearing (16) on to worm and wheel (19, 21).
18. Tachometer.
19. Worm.
20. Shaft emergency governor.
21. Worm wheel driving regulating governor and oil pump.
22. Pivoted thrust block adjusting gear.
23. Handwheel for operating (22).
24. Main oil pump (submerged in oil tank).
25. Oil tank.
26. Non-return valve.
27. Oil pump delivery pipe to oil cooler.
28. Steam-end pedestal.
29. Flexible half-coupling.
30. Generator bearing.
31. Oil baffle.
32. Cleading for turbine cylinder.

are required. This state of affairs arises from the following considerations—

Owing to the pressure drop over the rows of moving blades as well as over the rows of "guide blades," it is impossible to resort to partial steam admission (as in the case of impulse turbines), and consequently steam has to be admitted all round the shaft at every "stage" of the expansion. With high pressure steam, the initial specific volume is quite small, and consequently the areas to be provided for the passage of the steam through the blades at the inlet end are also comparatively small, with the result that, if the blades at the front end were placed on the same mean diameter as they are at the exhaust end, their height would be so small that a large proportion of the steam would leak over the blade tips, and a great loss of efficiency would result. The mean diameters are, therefore, greatly cut down at the front end of the turbine, in order that reasonable blade-heights may be permissible. The low mean-blade speed which results (at the inlet end) thus entails low steam velocities and correspondingly low rate of conversion of heat energy into mechanical work, and thus the turbine is inevitably lengthened. Balancing pistons are also required to balance the steam pressures on the shoulders of the shaft, and on the rings of moving blades.

Fig. 9 shows a typical "reaction steam turbine" design for land purposes (due to *Messrs. C. A. Parsons & Co., Ltd., of Newcastle-on-Tyne,* who were the first to adopt "pressure compounding"). Very high steam economy is the great characteristic of this type. The "reaction" principle gives the highest efficiencies yet attained in turbine design, and in large sizes the Parsons' single rotation parallel-flow reaction steam turbine holds the field in this respect. For smaller outputs, the "reaction" principle still gives the best results, but with a different design, which is illustrated on Fig. 10, and is a radial flow turbine developed in Great Britain by the *Brush Electrical Engineering Co., of Loughborough.* It will be seen that there are two discs, one carrying the concentric rings of "guide blades," and the other carrying the concentric rings of "moving blades." As the steam travels outwards from the centre of the discs, it traverses a ring of blades on each disc alternately, until finally it exhausts all round the periphery of the discs. Both discs, however, rotate at equal speeds, but in opposite directions. Each rotating element drives a separate shaft of its own, and thus if the plant is a turbo-alternator, there will be two alternators, electrically connected in parallel. The mean blade speed of a ring on one disc (or rotating element) relative to the corresponding ring on the other (rotating) element, is thus double what it would be in the case of an ordinary single rotation turbine, and consequently such a "reaction pair" on any given mean diameter is equivalent to four such pairs in a single rotation turbine, with the same speed of rotation as that of each element in the former case—since the heat drop varies as the square of the jet velocity to be generated. Very good



Horizontal centre line of turbine.

FIG. 10.—BRUSH-LJUNGSTRÖM COMPOUND REACTION STEAM TURBINE.

a, *a* = steam pipes to turbine
b = shaft gland
c = steam inlet into centre of blade rings.
d, *e* = discs carrying concentric blade rings.
f = last ring of blades
g = steam pressure balancing discs.

(*Messrs. The Brush Electrical Engineering Company, Ltd.* (*Loughborough*).

efficiencies indeed are obtained with this type of compound reaction turbine, and extreme ingenuity has been shown on the part of the engineers responsible for its design in overcoming the many problems of construction inherent in it, both as regards reliability and accessibility.(²) As in the case of the Parsons design, great care has to be taken to minimize the leakage of steam over the tips of the blade rings, which takes place owing to the drop in pressure over each ring, and the steam thrust on each disc has to be specially balanced.

THE SIMPLE "REACTION WHEEL" in case (2) has no commercial representative, owing to the fact that, as the velocity ratio and the efficiency tend to unity, the driving torque vanishes, and consequently a reasonable output cannot be obtained except at a poor efficiency. The historical jet wheel ascribed to Hero of Alexandria (about the year 130 B.C.) was an example of this type. "Barker's Mill" is an application of the principle to water turbines.

CURVES OF EFFICIENCY ATTAINABLE WITH THE DIFFERENT TYPES OF BLADING.

The curves shown in Fig. 11 represent the net efficiencies obtained with—

(1) A "three-row Curtis wheel" design;
(2) A "two-row Curtis wheel" design;
(3) A "single-row 'Rateau' wheel" design;
(4) A group of "reaction blading";

after eliminating all measurable losses.

They are plotted on a base of velocity ratio, and represent what is attainable in compound steam turbines, when the velocity of exit of the steam from one compartment is utilized in the next. In the case of the Curtis wheels, disc-friction and blade "ventilation" losses have been eliminated, whilst in the case of the "Rateau" single-row wheels, the results are corrected from wheels receiving steam from the nozzles all round the periphery (that is to say, without partial admission), and revolving in low-pressure steam, where the loss due to disc friction is very slight.

In the case of the "reaction blading," the leakage due to steam escaping over the tips of the blades has been eliminated. We thus have comparative curves of the inherent efficiencies attainable from the different types of blading.

It will be noticed that all the "velocity compounded" wheels give comparatively poor results. The single-row "Rateau" wheels are efficient, whilst the best performance with "Reaction blading" has the surprising value of 90 per cent.—a figure which has recently been considerably improved upon.

Equivalent Wheels. For a given mean blade speed, then the greater the number of rows on a "velocity compounded"

(²) *See* list of references at end of article.

wheel, the lower the velocity ratio need be in order to attain the point of maximum efficiency; but on the other hand, the greater the number of rows, the less is the actual value of that maximum efficiency. For a given heat drop, and mean blade speed, we may tabulate equivalent wheels approximately as follows—

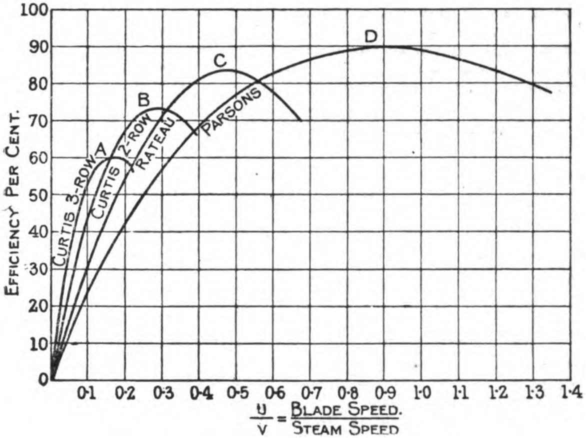


Fig. 11.—Comparative Efficiencies of Turbine Blading Systems.

One three-row wheel (Curtis);
Three two-row wheels (Curtis);
Twelve single-row wheels (Rateau);
Twenty-four "reaction pairs" (Parsons).

Then, if the steam conditions and speed of rotation are such that the three-row wheel is operating at the velocity ratio corresponding to the maximum efficiency, all the other combinations will also operate at their respective points of best efficiency, assuming that the heat drop is approximately equally divided in the case of the multiple wheels. But, although these several combinations are working at similar points on their respective efficiency curves, the actual efficiency values are very different.

We have seen that when steam turbines are "pressure compounded," the energy of expansion of the steam is divided up into smaller portions, so that the jet velocity in each "compartment" is greatly reduced, and consequently a much more moderate mean blade-speed may be adopted, and yet the best obtainable efficiency realized. This is the fundamental principle underlying the original Parsons turbine (1884).

The Method of Calculation of the overall velocity ratio for any given steam conditions, and for any given design, varies with different designers, but the following method has the advantage of universal application, and is very simply arrived at.

Let H_A be the total heat-drop due to the expansion of the steam right down to the vacuum pressure, then if the turbine has N compartments, or N moving rows in the case of "reaction" turbines, each working at the same velocity ratio, we have—

$$\text{velocity ratio} = \frac{\text{blade speed}}{\text{steam speed}} = \frac{U}{V}$$

$$= 0{\cdot}62\sqrt{\frac{K}{H_A}} \text{ for the pure pressure compounded impulse type} \qquad (2)$$

$$= 0{\cdot}87\sqrt{\frac{K}{H_A}} \text{ for the reaction type} \qquad (3)$$

Where K is a constant for a given design, and is equal to

$$S\,\{(\text{mean diameter in inches} \times \text{R.P.M.})^2 \times N \times 10^{-9}\} \qquad (4)$$

For a given heat drop and mean diameter per compartment the jet velocity produced in the guides of a reaction pair is only $1\sqrt{2}$ of that which would be developed in the nozzles of an impulse wheel, the velocity ratio thus being $\sqrt{2}$ higher than in the impulse case, because in the former case the heat drop is divided equally over the guide-row and moving row respectively. Thus, suppose that we have a compartment compounded impulse turbine with nine wheels, each 47 ins. diameter, running at 3,000 r.p.m. Let the steam conditions be 200 lbs. S.V.P. gauge, 200 deg. F. initial superheat, 195 lbs. per sq. inch absolute at entry to the turbine blading, and 28 ins. exhaust vacuum. The total available heat, from the "Mollier" diagram, is 379 B.Th.U. per lb., and we have

$$\text{Total } K = (47 \times 3{,}000)^2 \times 9 \times 10^{-9} = 179.$$

Hence the overall velocity ratio with which the turbine will operate is

$$\frac{U}{V} = 0{\cdot}62\sqrt{\frac{179}{379}} = 0{\cdot}426$$

In commercial engineering practice, a range of standard turbine frames is usually decided upon, and for any given operating conditions the frame most suitable is fixed upon, the blading then being made as far as practicable to suit those conditions. Thus, if the above turbine were adopted for steam conditions of, say, 180 lbs. S.V.P. gauge, 100 deg. F. initial superheat, 175 lbs. "absolute at the blading," and 28½ ins. vacuum, the total available heat would then be 364 B.Th.U.

per lb., and if the running speed were to be 3,000 r.p.m., the overall velocity ratio would be

$$0{\cdot}62\sqrt{\frac{179}{364}} = 0{\cdot}434$$

In the case of a reaction steam turbine, if the total K were 179, and the steam conditions were the same as in the first example of impulse design, viz., 200 lbs. S.V.P. 200 deg. F. initial superheat, 195 lbs. "absolute at the blading," and 28 ins. vacuum, the overall velocity ratio would be

$$\frac{U}{V} = 0{\cdot}87\sqrt{\frac{K}{H_A}} = 0{\cdot}87\sqrt{\frac{179}{379}} = 0{\cdot}60$$

Owing to the different shaft diameters used in any one "reaction" design, one row of blades at the exhaust end has the same K value as several rows at the inlet end. It is usual to choose the number of pairs of rows (one moving and one fixed) of any one blade height, and known as an "expansion," so that the total K is divided up approximately equally over each "expansion." Here K is the sum of the values of

$\{10^{-9} \times$ (mean diam. $\times$ revs.$)^2 \times$ (number of moving rows of the same height)$\}$ throughout the turbine.

It has been found from long experience that the most satisfactory distribution of K on a three-drum reaction turbine is ¼ on the first drum, ¼ on the second, and ½ on the third.

Choosing a Standard Frame to suit a given output and given steam conditions. In condensing turbines, the maximum attainable output is nearly always dependent on the permissible kinetic throw-away loss at the exhaust (known as the "leaving loss") due to the fact that the steam leaves the last row of moving blades and enters the exhaust with a certain velocity, and the energy thus carried away is a dead loss on the turbine. The greater the volume of steam we try to force through a given exhaust blade area, the greater its velocity will have to be in order to do it, and the greater the leaving loss.

Let D = the mean diameter of the blade-ring on the last wheel in inches.
h = the final blade-height in inches.
k = the effective opening per inch circumferentially, through the blades.
Q = the total steam consumption of the turbine in lbs./hr.
v = the specific volume of the steam at the exhaust pressure in cubic feet per lb.
V = the relative velocity of the steam through the exit of the blade passages, in feet per sec.

Then we have,

$$\frac{\pi D h k}{144} = \frac{Qv}{V \times 3600} \qquad (5)$$

Now the maximum permissible blade-height on a given mean diameter is usually fixed at one-fifth of that diameter, since if it is made more than this value, the "spread" of the moving blades towards their tips gives a bad shape to the steam passages there. Thus, we may write

$$\frac{\pi D^2 k}{144 \times 5} = \frac{Qv}{3600V} \qquad (6)$$

So that

$$V = \frac{Qv}{5\pi D^2 k} \text{ ft./sec.} = \text{the relative velocity of the steam through the blades} \qquad (7)$$

From a velocity-diagram (*cf.* Fig. 15), we can get V_o, the absolute velocity of discharge of the exhaust steam (see page 531). Thus, V_o can be either calculated by trigonometry, or graphically, the latter method being the more satisfactory, as it gives a visual indication of the state of affairs.

If h_a is the heat-equivalent of the kinetic energy thrown away at the exhaust, we have

$$\frac{V_o^2}{2gJ} = h_a \qquad (8)$$

So that the final "leaving loss" is $\left(\frac{h_a}{H_A} \times 100\right)$ per cent. of the total available heat.

It is not usual, in high-pressure condensing turbines, to permit h_a to exceed 5 per cent. of H_A, and this figure is only tolerated when the exhaust vacuum is very high, and the maximum possible output from a given frame is required.

Thus, assuming a given "leaving loss," it is a simple matter to work backwards in any given case, and find out what value of Q gives this percentage loss on the efficiency-ratio. The probable efficiency-ratio of a turbine is approximately known by experience, so that the approximate steam consumption of the turbine in lbs./k.w. hour will be known, and this divided into Q gives the approximate output in k.w.

Thus, 1 kilowatt-hour

$$= 3{,}412 \text{ B.Th.U., so that} \qquad (9)$$

$3{,}412/H_A$ = the theoretical steam-consumption of the turbine, in lbs. per kilowatt hour, and (10)

$\frac{3{,}412}{(\varepsilon)H_A}$ = the approximate actual steam consumption, whilst

$\frac{Q\varepsilon H_A}{3{,}412}$ = the approximate output in k.w. (11)

The maximum output from a given frame will be discussed later, in its relation to the permissible stresses in the turbine component parts.

THEORY OF STEAM TURBINE DESIGN

The Problem of Nozzle Design (see Fig. 12).

When steam expands adiabatically through an orifice, both the velocity and specific volume increase owing to the expansion.

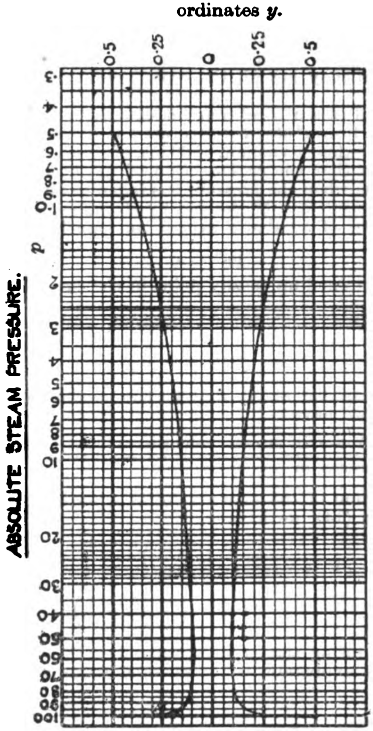


Fig. 12.—Theoretical Nozzle Profile for an Elastic Fluid.

At first, the velocity increases faster than the volume, so that a convergent passage is first required. A point is soon reached, however, where the unital increase in velocity is exactly equal to the unital increase in specific volume, so that there will be a point of minimum cross-section, called the "throat." On further expansion, however, the specific volume begins to increase faster than the velocity, so that a "diverging" mouthpiece or nozzle-exit will be required. The profile of the nozzle will, therefore, be as shown in Fig. 12.

Assuming a circular cross-section at every point, then the ordinates y_1 y_2, etc., represent the different diameters required at points in the expansion where the pressures are respectively P_1 P_2, etc. The abscissae are indeterminate as regards position in the nozzle, for it is clear that we may squeeze up or draw out the convergent inlet and divergent outlet as we please, provided that in practice the curvature is not made too steep, in which case the steam may not fill the cross-section of the nozzle at every point.

The assumed steam conditions in calculating the profile in Fig. 12 are 100 lbs./in² *abs.* dry-saturated steam at inlet, expanding down to an exhaust vacuum of 29 ins. of mercury (bar 30 ins.). It will be seen that the cross-section of the nozzle reaches its minimum value when the pressure has fallen to about 58 lbs./in² *abs.*, and this result can be verified by theory—

From the law of adiabatic expansion, namely,

$$Pv^n = C \tag{12}$$

the velocity attained up to any point in the expansion is given by the equation

$$V_2 = \sqrt{2g\left(\frac{n}{n-1}\right) P_1v_1 \left\{1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right\}} \tag{13}$$

Now it was reasoned above that at the smallest cross-section of the nozzle we have—

$$\frac{\text{unital increase in steam velocity}}{\text{unital increase in specific volume}} = \text{unity} = \frac{dV.}{dv.}\,\frac{v}{V} \tag{14}$$

By differentiation of the above velocity equation (13), and by substitution in (14), we have—

$$\frac{dV.}{dv.}\,\frac{v}{V} = \frac{(n-1)\left(\frac{v_1}{v_2}\right)^{n-1}}{2\left(1 - \left(\frac{v_1}{v_2}\right)^{n-1}\right)} \tag{15}$$

at any point during the expansion.

Now, if this expression is to be equal to unity, it is easily shown that

$$\frac{v_1}{v_2} = \left(\frac{2}{n+1}\right)^{\frac{1}{n-1}} \tag{16}$$

so that

$$\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \tag{17}$$

where P_2 is the pressure at the throat.

Thus there will always be a pressure at the throat, or point of minimum cross-section, bearing a definite relation to P_1, the initial pressure, provided that the exhaust pressure is equal to or less than this value. When this is the case, the pressure at the throat is called the "critical pressure," and careful consideration of what has just been said will show that the discharge through the nozzle is then a maximum, and should be unaffected by further reduction of the exhaust pressure, no matter to what extent.

For initially dry-saturated steam, equation (17) gives

$$P_2/P_1 = 0{\cdot}58$$

(n being taken as 1·135) for the "critical" pressure ratio, and the velocity at the throat is then

$$V_2 = \sqrt{gnP_2v_2} \qquad (18)$$

and remains unaltered when the exhaust pressure is further reduced.

NOZZLE "FLARE."

This denotes the ratio

$$\frac{\text{nozzle exit area}}{\text{nozzle throat area}} = \frac{A_a}{A_t} \qquad (19)$$

and is always greater than unity if the "critical" pressure ratio is exceeded. If it is not exceeded, then no mouth-piece at all is required, and we have a "simple convergent nozzle."

It can be shown that the amount of "flare" depends solely on the pressure-ratio, and is independent of the actual values of the pressure (3).

To continue the argument on the discharge through nozzles:—
When the pressure at the throat has its "critical" value, the equation of flow, that is—

$$Q = A_t\sqrt{2g\left(\frac{n}{(n-1)}\right)\frac{P_1}{v_1}\left\{\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right\}} \qquad (20)$$

then becomes

$$Q_{\max} = aA_t\sqrt{\frac{P_1}{v_1}} \qquad \left(\text{or} = bA_t\sqrt{P_1\left(\frac{n+1}{n}\right)}\right) \qquad (21)$$

Now, $P_1v_1 = H$, the "Homogeneous Head" of the steam in its initial condition. Hence we may write—

$$Q_{\max} = \frac{P_1A_t}{c}$$

where c is a coefficient, or

$$\frac{Q}{P_1} = \frac{A_t}{c} \qquad (22)$$

(3) *See* list of references at end of article.

thus c depends on the initial steam conditions, and we have—

$$\frac{1}{c} = \sqrt{\frac{2g}{\mathrm{H}}\left(\frac{n}{n-1}\right)\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}\left\{1-\left(\frac{2}{n+1}\right)^{\frac{n+1}{2}}\right\}} \qquad (23)$$

but in practice it is better to determine c from experimental data and from the "Mollier" diagram.

Formula 22 is known as "Napier's Formula." When we take for our units

Q = lbs. of steam per hour passed by the nozzle,

A_t = nozzle throat area in square inches,

P_1 = initial steam pressure in lbs. per square inch (*abs*) at the entry to the nozzle,

Then we have

$$\frac{\mathrm{Q}/3600}{\mathrm{P}_1} = \frac{\mathrm{A}_t}{c}$$

For steam, the value of the coefficient c may be taken approximately as

75 for high-pressure steam, initially highly superheated,

70 for high pressure initially dry saturated steam, and

65 for steam at atmospheric pressure.

These figures are sufficiently accurate for running out a preliminary calculation of a nozzle design.

A satisfactory way of finding c for design purposes is to combine equation 22 with the equation of flow

$$\mathrm{Q}v = \mathrm{AV}$$

and obtain

$$c = 144\ \mathrm{P}_1 \frac{v_t}{\mathrm{V}_t} \qquad (24)$$

where v_t and V_t are specific volume and jet velocity respectively, at the "throat."

In this equation the initial state of the steam is assumed to be known, and having located it on the "Mollier" diagram, we can draw the expansion line for the nozzle (usually assumed to be adiabatic, since the nozzle efficiency is high), and thus obtain the velocity and specific volume at the throat by marking on the expansion line the point where the pressure has fallen to the "critical" value given by equation (17).

The heat drop up to that point, and the quality at that point, may then be read off.

Napier's Formula only applies to cases where the "critical pressure ratio" is reached at the nozzle throat, such as in single wheel turbines like the De Laval, or the Curtis turbines, and some designs of Rateau turbines. As before stated, if the expansion ratio over the nozzle is in excess of the value given by the relation

$$P_2/P_1 = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

then an additional "divergent' mouthpiece to the nozzle becomes necessary, although the discharge is unaltered.

If this divergent mouthpiece is not fitted, and the exhaust pressure of the wheel were below the "critical" value, then the "critical pressure" would still exist at the throat, and further expansion (i.e. further increase in velocity) would take place out in exhaust space. The expansion would, however, be unstable, and the final velocity of the jet would fall much below the theoretical value, on account of eddies.

In practice, for parallel flow turbines, the nozzle has to be "skewed" at an angle with the plane of the wheel, so that the steam may enter the blades, but this does not invalidate the foregoing discussion.

As a matter of fact, it is found that, even with a correctly proportioned divergent nozzle, some expansion does take place outside the nozzles, and in the blades themselves, and this has the effect of making the designed nozzle mouth-divergence too great. Experiment shows that over-divergence produces more serious losses in the jet-efficiency than under divergence, and for this reason it is better to underflare the nozzle when designing it. The taper of the sides of the diverging mouthpiece in any case should not be made too steep, or the rapidly moving steam will not fill it, with the result that eddies and consequent inefficiency will be produced.

Having thus dealt shortly with the theory of convergent-divergent nozzles, we must now shortly review the problem of

IMPULSE BLADING DESIGN.

Turning back to page 513, we recall that in the theory of this type of turbine, all the conversion of heat to kinetic energy takes place in the nozzles, and none in the blades. The specific volume of the steam passing through the blade passages remains practically constant, but in the revolving blades the steam velocity steadily diminishes as work is done upon the wheels. We have, therefore, to find some method of taking this into account when proportioning the blades.

When two or more rows of revolving blades are used per wheel, as in the Curtis arrangement, it is usual to make the blade-exit angles less than the corresponding inlet angles (*see* velocity diagrams, page 534), otherwise the steam jets from the stationary nozzles and guide-blades may strike the backs of the revolving blades and exert a certain amount of retarding influence on them, thus diminishing the effective driving torque on the wheels. Further, in order to prevent "spilling," it is necessary to make the first row of revolving blades overlap the nozzles, and each successive row of blades to overlap its predecessor. This leads to the method now universally adopted, i.e. of increasing the blade-heights from row to row. We then get the arrangement shown in Fig. 13.

To assist in preventing "spilling" of the steam over the blade-tops, they are "shrouded."

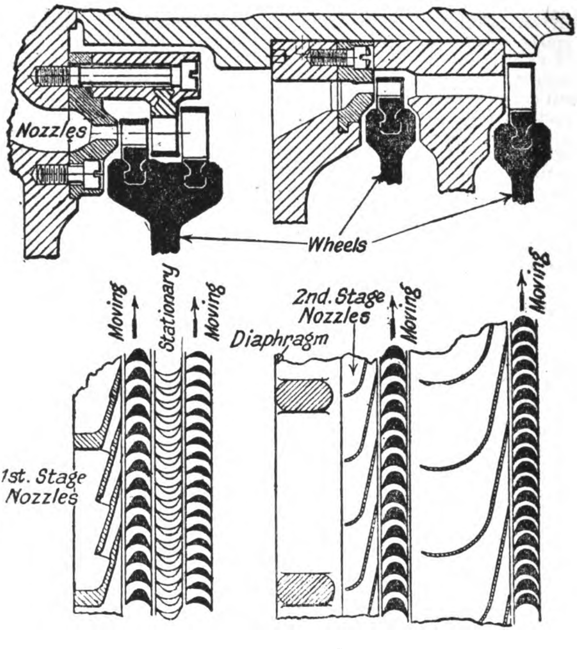


FIG. 13.—CURTIS BLADING.
(*The British Thomson-Houston Co., Ltd., Rugby.*)

THE VELOCITY DIAGRAM AND THE PRINCIPLE OF "RELATIVE" VELOCITY.

If we have a bucket B (Fig. 14A), moving in a direction BU with a velocity U, and a particle of steam A moving in the same direction with a velocity V, which is greater than U, then it is evident that the particle A will overtake the blade, but not so quickly as it would do if the latter were stationary. If V = U, then A will never overtake B, and the two will always remain the same distance apart. When the blade B is stationary, the velocity of A relative to it is V; when the blade is moving at the same speed as A, the velocity of A relative to B is zero. Taking an intermediate case, it is clear that if A is moving faster than B, it will overtake it with a velocity V − U. In other words, the velocity of A relative to B is always V − U. This state of affairs is graphically represented by the vector diagram in Fig. 14B where AB = V, AC = U, so that BC = V − U.

Now suppose that the steam particle is not moving quite in the same direction as the blade, but at an angle to it. Then, applying the same principle of vector subtraction, we make AB = V, and then the line joining B and C gives the direction and magnitude of the velocity of A relative to B. This is precisely what occurs in a steam turbine.

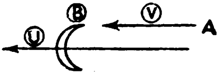


FIG. 14A.

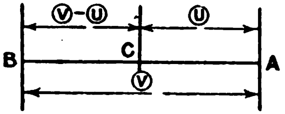


FIG. 14B.

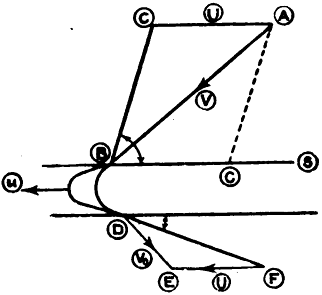


FIG. 15.
THE VELOCITY DIAGRAMS.

In Fig. 15 we have the nozzle directing a steam jet AB on to the moving row of blades B. Then AC = U, the mean blade-velocity, and BC represents in direction and magnitude the velocity of the particles of the steam jet relative to the blades. Clearly, then, in order that the steam may enter the blade channels smoothly, and without striking the backs of the blades, the inlet angle of the blades must not be less than the angle CBS.

The steam travels over the blade surfaces with constant velocity BC, doing work in driving the wheel, and its final absolute velocity of exit (relative to the ground) is much reduced. To find it, we repeat the process, and, as before, set off DF = BC,

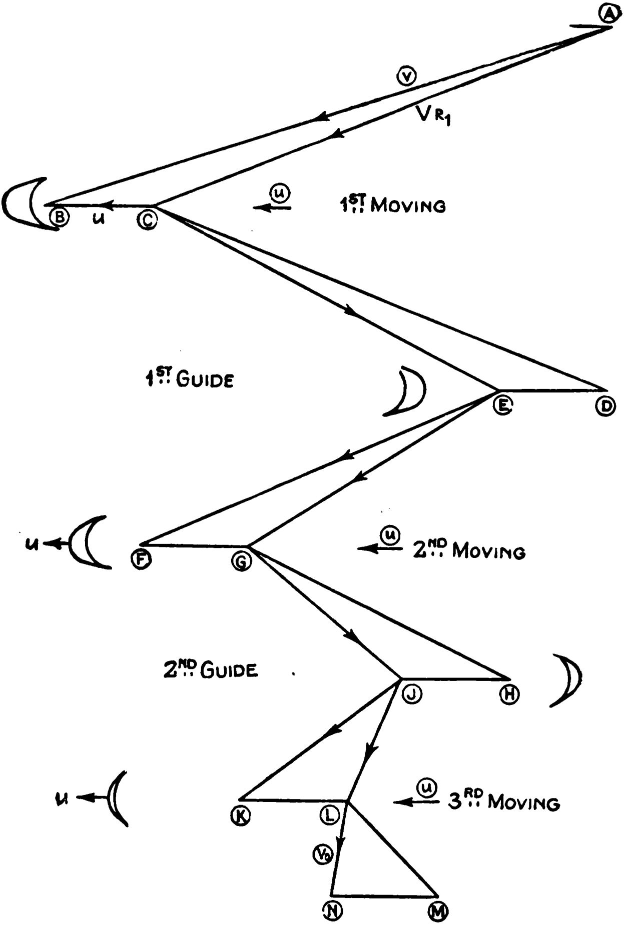


FIG. 16.—THREE-ROW WHEEL VELOCITY DIAGRAM.

at an angle equal to the blade exit angle, and from F set off FE equal and parallel to U, the mean blade velocity. Then DE represents in direction and magnitude the absolute velocity of exit from the moving blades.

If the wheel is a simple Rateau, this ends the process in the first compartment, and it is repeated in the remaining compartments; but if it is of the Curtis "velocity compounded" type we merely continue the process, as shown in Fig. 16, until we arrive at the final exhaust velocity from the wheel, LN. A more convenient way of drawing this straggling velocity diagram is shown in Fig. 17, in which the letters correspond to the previous diagram, and are self-explanatory.

Now the quantity of steam passing per second longitudinally down the turbine at any point is the same and therefore we must have—

(longitudinal velocity × area of annulus) = (lbs. of steam per second × specific volume),

$$\text{i.e.}\left(V \times \frac{h'' \times l''}{144}\right) = \frac{Q \times v}{3600}$$

where h = blade height
and l = arc of steam admission,

so that $$h = \frac{Qv}{25Vl} \text{ inches} \qquad (25)$$

Now if all the expansion of the steam takes place in the nozzles, and none in the blades, the specific volume of the steam passing through the blades must (neglecting friction) remain constant. Also for any particular wheel, Q is constant, also l, the arc of steam admission. Therefore h, the blade height, is inversely proportional to the longitudinal velocity V.

Hence the calculation of the impulse blade proportions proceeds as follows (the method has been developed by Professor G. Stoney, F.R.S.)—

let $X = P_1/P_2 = 180/60 = 3{\cdot}0$ = expansion-ratio,
then H_A = available heat drop = 87 B.Th.U./lb., so that the initial jet velocity $= 224\sqrt{87} = 2{,}080$ ft./sec.
let U = mean blade velocity = RD/229 = 3,000 × 35/229 = 460 ft./sec., and we have (Fig. 18).

	Nozzle.	Blade.
Assumed exit angle . . .	18°	18°
Diagram inlet ,, . . .		22½°
Outside angle (allow 4½° for entry) .		45°
Longitudinal velocity at exit .	630	510
Assumed nozzle height . . .	½″	
Required blade exit height . .	$\frac{1}{2} \times \frac{630}{510} = 0{\cdot}618''$	

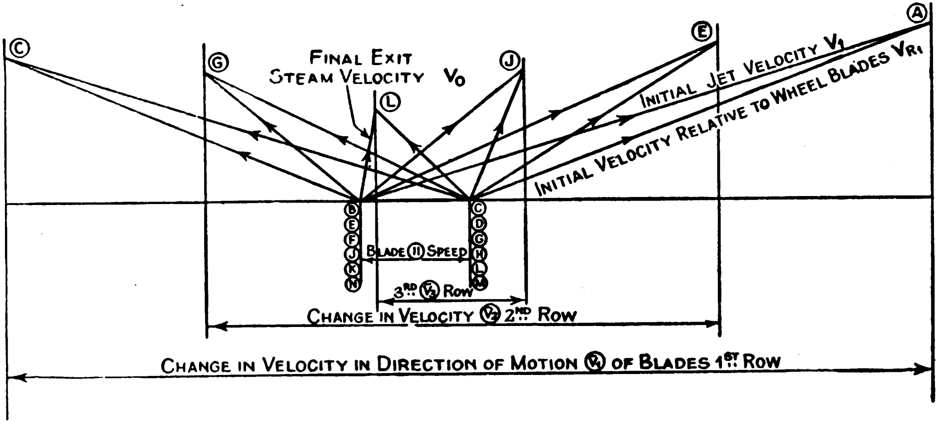

Ⓐ
Ⓔ
Ⓙ
Ⓛ
Ⓖ
Ⓒ
Final Exit Steam Velocity V_0
Initial Jet Velocity V_1
Initial Velocity Relative to Wheel Blades V_{R1}
Ⓑ Ⓔ Ⓕ Ⓙ Ⓚ Ⓝ
Ⓒ Ⓓ Ⓖ Ⓗ Ⓛ Ⓜ
Blade ⑪ Speed
3rd V_3 Row
Change in Velocity V_2 2nd Row
Change in Velocity in Direction of Motion V_1 of Blades 1st Row

FIG. 17.—CONVENIENT METHOD OF DRAWING VELOCITY DIAGRAM FOR CURTIS VELOCITY WHEEL.

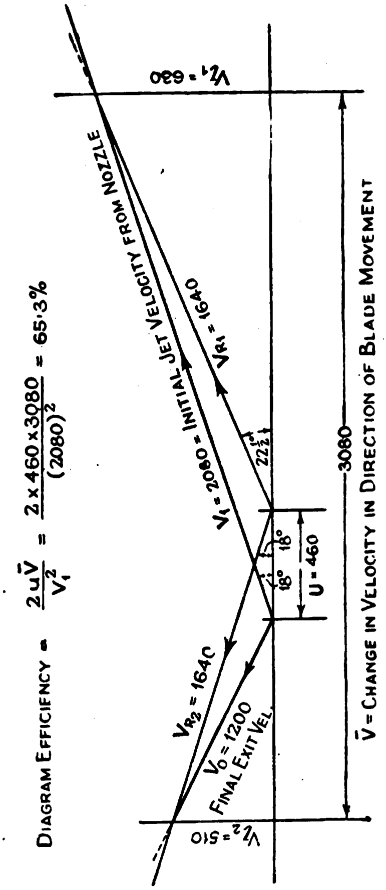


Diagram Efficiency = $\frac{2u\bar{V}}{V_1^2} = \frac{2 \times 460 \times 3080}{(2080)^2} = 65{\cdot}3\%$
$V_{Z_1} = 680$
Initial Jet Velocity from Nozzle
$V_1 = 2080$
$V_{R_1} = 1640$
$22\frac{1}{2}°$
18°
18°
U = 460
$V_{R_2} = 1640$
$V_0 = 1200$
Final Exit Vel.
$V_{Z_2} = 510$
3080
$\bar{V}$ = Change in Velocity in Direction of Blade Movement

Fig. 18.—Application of Velocity Diagram to design of Blading.

The process is merely extended if there is more than one row of moving blades. In the above case, it appears that we want a blade with an "outside angle" of 45 deg., when we assume that it should be set with an exit angle of 18 deg. Since, however, the blades used in practice have "tails" of finite thickness, a correction to the blade height will be necessary in order to take this into account.

THE DIAGRAM EFFICIENCY η.

The "diagram efficiency" of the wheel, that is the efficiency calculated from the diagram, is equal to—

$$\eta = \frac{\text{work done per second}}{\text{energy in steam jet}}$$

$$= \frac{\text{force on blades} \times \text{distance travelled per unit time}}{\text{energy in steam jet}}$$

$$= \frac{\text{change of momentum per sec.} \times \text{mean blade-velocity}}{\text{energy in steam jet}}$$

$$= \frac{\dfrac{\text{WU}\,(\overline{\text{V}_1} + \overline{\text{V}_2})}{g}}{\dfrac{\text{WV}_1^2}{2g}} \quad \text{for a "two-row wheel"}$$

$$= \frac{2\text{U}(\overline{\text{V}_1} + \overline{\text{V}_2})}{\text{V}_1^2} \qquad (26)$$

and so on for three-row and four-row wheels.

Now, in actual practice, it is found that considerable modifications have to be made to this method, in order to make it applicable to actual design, but this does not alter the principle.

In the first place, if we take, say, a two-row wheel of given mean diameter, fix the steam jet velocity, run the wheel at different speeds of rotation, find the speed that gives the highest efficiency, and then try and draw the velocity diagram to suit, we may get into difficulties.

It, may possibly be found that the magnitude of the velocity of the steam jet at the entry to the second moving row, is so small that the absolute exit velocity (as found by the diagram), has a direction the same as that of the blades, so that the second row appears to be doing practically no work, whereas if we were to remove the second row of blades, and run the wheel at what was formerly the best speed, the efficiency might be found to be less than before. The principal reasons causing this result are—

(1) The initial jet velocity is not the theoretical due to the heat-drop, owing to friction and eddy losses in the nozzle.

(2) The initial jet velocity is not that due to the theoretical heat-drop, because the heat-drop does not all take place in the nozzles, but a small part in the blades themselves.

(3) The specific volume of the steam as it passes through the blades is not constant, on account of the expansion that takes place in the blades themselves, and on account of the heating

up of the steam or drying due to friction and eddies in the blade-passages themselves.

(4) The velocity of the steam through the blade-passages is increased by the expansion that takes place in them, but diminished owing to the friction of the blade surfaces.

All these different effects alter the behaviour of the steam in the impulse wheel, and the best design of nozzles and blading will be made by the engineer who possesses the greatest knowledge, skill and judgment in so adjusting his method of drawing the velocity-diagram that in practice the actual velocities of the steam are what the diagram leads him to think they are. He runs a series of experiments with carefully designed impulse wheels, finds out in which direction to vary the proportions and thus gradually converges on the best arrangement by continually comparing his experimental results with what is theoretically attainable.

Looking at the problem from the mechanical point of view, it may be noted that the comparatively small pressure-drop over the moving blades of well-designed impulse turbines does away with the necessity for fine radial working-clearances, and these are commonly made $\frac{3}{16}$ in. or more. It is found, however, that in order to attain the highest efficiency, the axial clearances between the nozzles and blading must be minimized. This is a circumstance which has to be very carefully watched in practice, especially in large turbines, as it involves running a heavy impulse wheel close up (at the rim) to a massive diaphragm. In other words, it is not sufficient that an impulse wheel should be designed solely from the point of view of withstanding centrifugal forces; it must also be made rigid enough to revolve without "flutter" at the rim, so that it can be run with safety close up to the diaphragm.

THEORY OF PRESSURE COMPOUNDED TURBINES.

We must now consider shortly the effect, in a pressure compounded turbine, of the friction losses of one wheel on the next. Turning to the Mollier diagram, let AB, in Fig. 19, be the pressure line corresponding to the stop valve pressure, and BC the superheat line for the value of the initial superheat. Say the initial steam conditions are 180 lbs. gauge S.V.P. and 100 deg. F. initial superheat. Let DE be the pressure line corresponding to the pressure after the governor valve and in front of the nozzles. Then the steam will be throttled down through the steam chest, without any external work being done. The internal energy therefore remains constant, that is the total heat remains constant, and thus BD represents the line of throttling. The superheat at D will be higher than at B, but the total temperature will be practically unchanged (Joule's Law).

Now we know that the vertical line between the pressure limits gives the theoretically-available heat per pound of steam for any initial state-point, such as B. Hence BN represents the maximum available heat per pound, for the case in question, and we see at once from the diagram how wasteful throttling is,

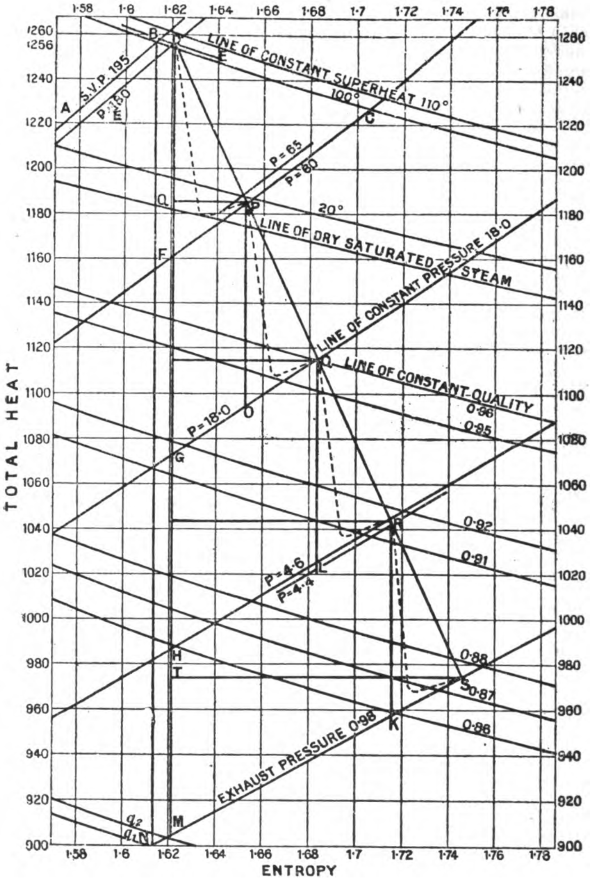


FIG. 19.—THE USE OF THE "MOLLIER" TOTAL-HEAT-ENTROPY DIAGRAM.

for the line DM is clearly shorter than the line BN, showing that although the initial total heat is unchanged by throttling, yet much less of it can be converted into work in a heat engine, and as a consequence more heat is thrown away to the exhaust. The exhaust steam is thus drier, the quality having increased from q_1 to q_2.

Suppose now that we are designing a Curtis turbine with four pressure-stages, each stage to be arranged to do equal work. Let the Available Heat drop DM be divided up into four equal parts, DF, FG, GH, and HM respectively. Then DF is the heat drop over the first wheel, and F lies on the pressure-line orresponding to the exhaust pressure to the first wheel (and at he same time to the inlet pressure to the second wheel). Then the exhaust passages to the first wheel are, at one and the same time, the inlet nozzles to the second wheel, so that the residual heat energy and velocity energy from any one wheel passes on to the next wheel, and so on. But if the passages are badly arranged, or are of a bad shape, the greater proportion of the residual velocity energy will be re-converted into heat energy. This has the same effect as the throttling at the inlet to the turbine, the steam is dried or superheated, and more heat is thrown away in the exhaust from that wheel, with consequent loss in economy. In fact, if the efficiency of the wheel be e, then if $DF = h_a$, the available heat in that compartment, a quantity of heat equal to eh_a is converted into work, and $h_a - eh_a$, or $(1 - e)h_a$ is re-converted back into heat-energy. The final condition of the steam at exhaust to that wheel must, therefore, lie at the intersection of the exhaust pressure line FP and the total heat line aP, where

$$Da = eh_a.$$

P is therefore the state-point at entry to the next wheel. This condition of the steam is better than we would have had at F, had the steam expanded without loss, that is to say adiabatically, and we should therefore expect the next wheel to have more heat available for work on this account.

This is precisely what occurs, and we see that PO is greater than FG, and similarly in the other wheel compartments,

$$QL > GH,$$
$$RK > HM.$$

The result is, therefore, that the total available heat in all the wheels is DF + PO + QL + RK, and the total of these is greater than DM, the adiabatic available heat from P_1 to P_e, although not so much of it is converted into useful work.

The overall blading efficiency is thus

$$\frac{e\,(DF + PO + QL + RK)}{DM}$$

and the ratio

$$\frac{(DF + PO + QL + RK)}{DM}$$

is called the "Reheat Factor."

It is a coefficient showing the slight gain in available heat due to the fact that each wheel receives its steam in slightly better condition than it would do were the expansion to take place straight down the line DM.

Of course, eRH_A can never be greater than H_A, and is only equal to it when $e = 1$ and $R = 1$, which is the ideal case of adiabatic (that is isentropic) expansion.

The important point to notice, however, is that the overall efficiency of the blading (= DT/DM) is greater than the "stage" efficiency e, in a compound steam turbine.

The value of R, the "Reheat Factor," for expansion between given pressure limits, can be approximately calculated when we assume that expansion takes place by an infinite number of indefinitely small stages, that is in an imaginary turbine consisting of an infinite number of simple turbines in series. The steps are then so small that DPQRS becomes the actual expansion line, instead of the stepped line, like DFPOQLRKS, and the "Reheat Factor" is given by the relation (4).

$$R = \frac{1 - \left(\frac{T_2}{T_1}\right)^e}{e\left(1 - \frac{T_2}{T_1}\right)} \qquad (27)$$

where T_1 and T_2 are the initial and final temperatures of the steam, whilst e is the "stage" efficiency (of each impulse wheel or pair of rows of reaction blading).

Using the relations

$$\left.\begin{array}{ll} & Pv^n = C \\ \text{and} & Pv = CT \end{array}\right\} \text{for superheated steam,}$$

we can convert the expression for R into the form(5)

$$R = \frac{1 - \left(\frac{P_2}{P_1}\right)^{e\left(1 - \frac{1}{n}\right)}}{e\left(1 - \left(\frac{P_2}{P_1}\right)^{1 - \frac{1}{n}}\right)} \qquad (28)$$

which gives the "Reheat Factor" in terms of the pressure-ratio for any given case, and is somewhat more convenient for design purposes.

Fig. 20 gives the values for R for different values of the stage-efficiency and pressure-ratio, calculated from equation (27).

(4) *See* list of references at end of article.
(5) *See* list of references at end of article.

$$R = \frac{1 - \left(\frac{T_2}{T_1}\right)^e}{e\left(1 - \frac{T_2}{T_1}\right)}$$

Values of exhaust pressures (Absolute).	Values of stage-efficiency "e."								
Pa bs.	0·1	0·2	0·3	0·4	0·5	0·6	0·7	0·8	0·9
300	1·0075	1·0066	1·0058	1·0049	1·0041	1·0032	1·0023	1·0015	1·0006
250	1·0161	1·0143	1·0124	1·0106	1·008	1·0069	1·0051	1·0032	1·0019
200	1·0265	1·0235	1·0205	1·0174	1·0144	1·0114	1·0083	1·0053	1·0029
150	1·0394	1·0349	1·0304	1·0259	1·0214	1·0169	1·0124	1·0079	1·0042
100	1·0569	1·0504	1·0439	1·0374	1·0309	1·0244	1·0179	1·0114	1·0061
70	1·0717	1·0635	1·0553	1·0471	1·0389	1·0307	1·0225	1·0143	1·0076
40	1·0936	1·0829	1·0722	1·0615	1·0508	1·0401	1·0294	1·0187	1·0099
30	1·1047	1·0927	1·0808	1·0688	1·0568	1·0449	1·0329	1·0209	1·0115
20	1·1191	1·1055	1·0919	1·0783	1·0646	1·0510	1·0374	1·0238	1·0125
15	1·1292	1·1145	1·0997	1·0849	1·0702	1·0554	1·0406	1·0258	1·0135
10	1·1429	1·1265	1·1102	1·0939	1·0775	1·0612	1·0449	1·0285	1·0149
7	1·1545	1·1369	1·1192	1·1016	1·0839	1·0663	1·0486	1·0309	1·0160
5	1·1651	1·1462	1·1274	1·1085	1·0896	1·0708	1·0519	1·0330	1·0170
3	1·1808	1·1601	1·1395	1·1188	1·0981	1·0775	1·0568	1·0361	1·0188
2	1·1926	1·1706	1·1486	1·1266	1·1045	1·0825	1·0605	1·0385	1·0198
1·5	1·2012	1·1782	1·1552	1·1322	1·1092	1·0862	1·0632	1·0402	1·0206
1·0	1·2120	1·1878	1·1636	1·1394	1·1151	1·0909	1·0667	1·0424	1·0217
0·5	1·2309	1·2045	1·1785	1·1417	1·253	1·0989	1·0725	1·0460	1·0234

Values of Reheat factor R for different exhaust pressures and stage efficiencies.

FIG. 20.—REHEAT FACTORS (R) FOR SATURATED STEAM, CALCULATED FROM 350 LBS. ABSOLUTE.

Example. What is the Reheat Factor (R) at 0·80 stage efficiency, for expansion from 250 lbs. abs. down to 0·5 lbs. abs. ?

For 350 lbs. abs. to 0·5 lbs. abs. R = 1·046
" " " 250 " " R = 1·0032

Difference 0·0428
+ 1

Required Reheat Factor = 1·0428

Empirical rule. $\left(R_{P_1P_2} - 1\right) + \left(R_{P_2P_3} - 1\right) = \left(R_{P_1P_3} - 1\right)$

or $R_{P_2P_3} = \left(R_{P_1P_3} - R_{P_1P_2}\right) + 1.$

The proof of this relation (27) for R depends upon the assumption that the equi-pressure lines in the "Mollier" diagram (which are straight lines in the saturation field) meet at one point away to the left of the bottom corner of the diagram. This assumption is a justifiable approximation to the truth, inasmuch as the intersections of the equi-pressure lines are very far to the left, and the angle between any two of the lines used in practice is very small.

In the superheat field, the equi-pressure lines are considerably curved, owing to the varying specific heat of superheated steam, but the "Reheat Factor" calculated for such conditions, by a step-by-step process on the chart, varies but little from that calculated by the formula (28).

It will be clear that if we are to proportion the nozzle areas correctly, especially for stages towards the exhaust end of the turbine, it is essential to know the shape and position of the expansion line all the way down. In actual practice, there is frictional loss in the nozzles themselves, and bearing in mind what was said in the discussion of velocity diagrams, we see that the actual stepped expansion line will be somewhat as shown in Fig. 19 by the dotted line.

An accurate collection of data on the properties of steam will also be essential, and in this relation Callendar's work should always be consulted, in spite of the fact that this book does not present the properties of steam in quite so convenient a form as other works on the subject.

It will be clear that great experimental knowledge, experience, judgment, and skill are necessary in order to design the best possible arrangement, and it is here that the individuality of the engineer is of account.

REACTION STEAM TURBINE DESIGN.

Hitherto, we have been dealing with turbines in which in each stage the "critical" pressure-ratio is exceeded, and the nozzle discharge is therefore unaffected by the exhaust pressure to that stage.

This is not the case in reaction turbines, nor, in many cases, in pure pressure compounded impulse turbines.

There is, however, no absolute necessity for this difference; that is to say the reason why the critical pressure-ratio is not exceeded in the cases mentioned, is that in order to run each simple turbine of the series at the necessary high velocity ratio, (*see* page 521), with the mean blade speeds that are possible in practice, the heat-drop has to be divided up into such small parts that the pressure-ratio over each does not exceed the critical value. It would not invalidate the reaction principle (in the case of reaction turbines) if it did. The advantage, however, that accrues to reaction turbines is as follows—

There are serious losses in the moving blades of impulse turbines, more serious than, and in addition to, the losses in the fixed nozzles. Reaction blades, on the other hand, at their best velocity-ratio are little more than fixed and moving nozzles alternately, and are subject to the latter losses only.

Let us now consider the nozzle (or blade) design of a reaction turbine of the usual type, that is to say where the critical pressure ratio at each row of guides or moving blades is not exceeded. As a consequence, therefore, the exhaust pressure from one row affects the steam discharge from that row, and it must be taken into account when proportioning the blading for a given steam discharge.

Just as the modern "reaction" steam turbine owes its inception to the analogy of the water turbine (which had already been brought to a high pitch of efficiency before the introduction of the steam turbine in a practical form in 1884), so the best way to design it is by analogy with the method used in the case of water turbines ([6]). When the amount of expansion in each row is very small, the change in specific volume is insignificant, and we may regard the flow of steam through the blade passages as being similar to that of water. Now, the fundamental "Bernouilli" equation for the flow of water is

$$H_1 + \frac{P_1}{\rho} + \frac{V_1^2}{2g} = H_2 + \frac{P_2}{\rho} + \frac{V_2^2}{2g} \qquad (29)$$

if there is "stream line" motion without friction or eddying of any kind. This equation simply says that, under these conditions, the total energy in the water (potential and kinetic) remains constant.

In the case of the nozzles of an actual steam turbine, the static head due to the steam in the steam-pipe is negligible, and if we also assume, for the sake of simplicity, that the initial jet-velocity in front of the fixed blading is zero, then the above equation simplifies down to

$$\frac{V^2}{2g} = \frac{P_1 \cdot P_2}{\rho} = \frac{dP}{\rho} = \frac{H\,dP}{P} \text{ where } H = 144\ Pv \qquad (30)$$

([6]) *See* list of references at end of article.

Thus, if we take, as an example, a turbine with blading on a constant mean diameter, then the mean blade speed will be the same for every row of blades. In order, therefore, to obtain the same "velocity ratio" throughout, it will be necessary to make the velocity of the steam jets constant throughout.

Hence, from (30), if V is to be a constant for every row, then $H\frac{dP}{P}$ must be constant. Now if we assume that expanding steam approximately obeys Boyle's Law ($Pv = C$), then H may be taken as constant, and the constant amount of expansion that must take place over every row of blades is $\frac{dP}{P}$. Hence, every row of blades (both moving and fixed) must be made higher than the one before, by an amount $\frac{dP}{P}$. Thus the blade heights are in geometrical progression, and the curve of blade heights will be a logarithmic one.

To make all the blade rings on the same mean diameter would, however, lead to very small heights at the high pressure end of the turbine, and very large heights at the exhaust end, if the blade discharge areas are to be correct throughout.

There are two methods of overcoming this difficulty—

(1) Stepping the turbine rotor down towards the inlet end;

(2) Making the discharge angle of the last few rows successively larger.

Method (1) enables reasonably large blade heights to be used at the inlet end (where the specific volume of the steam is comparatively small).

Method (2) enables the blade heights at the exhaust end to be cut down to reasonable limits (*see* page 524). It will be noted that the blade height on any given mean diameter depends upon the square of that diameter. For if the mean diameter be doubled, the mean blade speed will be doubled, so that the steam velocity must be doubled. But if the same blade heights were used on this mean diameter, the blade discharge area would be also doubled, whereas, to double the steam velocity we only require one half of the original blade area (i.e. of the area that would be required if the mean diameter had not been altered). Hence, we only require one quarter of the blade height that would be necessary on the smaller mean diameter.

In reaction turbines, owing to the comparatively small expansion that takes place at each stage, it is usual to make the fixed and moving row of each pair the same height, and, except when dealing with the low-pressure steam, to make several pairs of successive rows all of the same height. This is simply to aid commercial production of the blading.

The principles laid down in the foregoing discussion are graphically represented in Figs. 21 to 25. In Fig. 21, curve AAA gives the blade heights plotted on a base of total number of reaction pairs, for a high-pressure reaction turbine whose initial mean diameter is 30 ins. If this mean diameter be

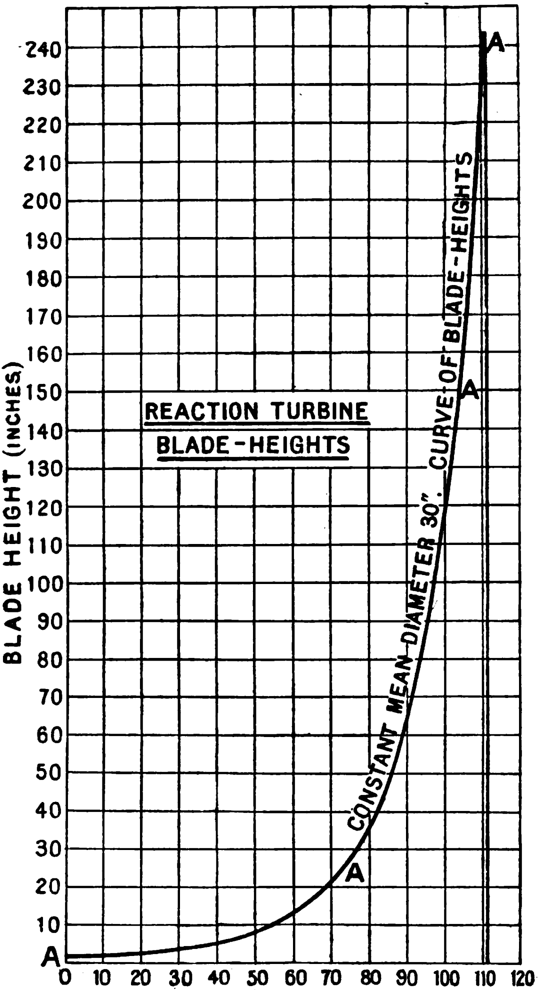

REACTION TURBINE
BLADE-HEIGHTS
CONSTANT MEAN DIAMETER 30". CURVE OF BLADE-HEIGHTS
A
A
A
A
BLADE HEIGHT (INCHES)
240
230
220
210
200
190
180
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
10
20
30
40
50
60
70
80
90
100
110
120
PAIRS OF ROWS

FIG. 21.

adhered to throughout, then, in order to efficiently expand the steam, we shall require about 111 reaction pairs of rows, and the blade height of the last moving row (assuming ' normal " discharge angle of about 19 deg.) works out to about 250 in.—a figure quite impossible of attainment on such a mean diameter, and quite impracticable also.

To overcome this difficulty, we resort to method (1) on page 544, but in the converse sense. That is, the mean diameter is increased (in steps, to aid commercial production)—and we then get a curve of blade heights shown by the stepped line ABCDE. Not only are the blade heights on the three larger mean diameters greatly reduced by this artifice, but the total number of reaction pairs is also greatly diminished, because the mean blade speed—or in other words, the K value—is much increased. Thus, only 54 pairs of rows are now required, where formerly 111 pairs were necessary. (Fig. 22.)

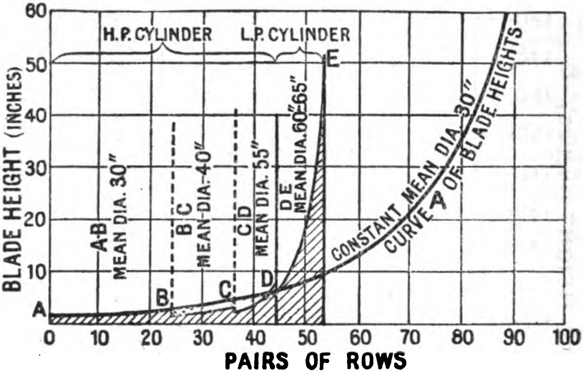


Fig. 22.—Reaction Turbine Blade-heights.

The blade heights at the exhaust end are, however, still impracticable, 53 ins. being now the maximum height.

In large turbines—and the blading that we are now considering concerns such—it is commercially possible to split up the expansion of the steam over two separate cylinders, a high-pressure and a low-pressure. This enables us to make the low-pressure turbine double-ended, so that the steam is divided up into two equal portions after leaving the high-pressure cylinder, and flows through the two low-pressure cylinders in parallel.

It should be noted in passing, that any given pound of steam can only flow through *one* low-pressure turbine, so that the double-ending of the low-pressure cylinder does not give any extra K value.

If the L.P. turbine be made double-ended in this manner, we are now able to halve the blade-heights in that cylinder, and thus arrive at the arrangement in figure 23. The last pair of rows is now diminished in height to 26½ ins., but the latter is still too large, for we saw on page 524 that for practical design it is not advisable to make the maximum blade-height more than one-fifth of the mean diameter. The maximum mean-diameter that we have adopted in this case (and which would be determined by reference to the speed of rotation and consequent rotational stresses that are involved) is 65 ins., so that the maximum blade-height should not exceed 13 ins.

We therefore have recourse to the final artifice (No. 2 on page 544) of making the discharge angle of the last few rows of blades progressively larger. Thus, if the discharge angle of the last row be made 45 deg. instead of the "normal" 19 deg., the capacity of the row (for a given blade-height) is about doubled, so that we can replace the last row of 26½ in. height by an equivalent row only 13 ins. in height, in both low-pressure turbines. The blading then takes the form indicated in Fig. 24.

Finally, in Fig. 25, the blade-heights are shown in relation to the mean-diameters for which they are designed, and the turbine can be designed as a whole, on this basis.

Now, we saw on page 522, that the mean overall velocity-ratio with which a turbine operates (when the velocity-ratios for all the stages are approximately equal) is given by the relation

$$a = \text{velocity-ratio} = 0{\cdot}87\sqrt{\frac{\text{total K}}{\mathrm{H_A}}} = a\sqrt{\frac{\mathrm{K}}{\mathrm{H}_a}}$$

Further, if we assume (as a first approximation) that expanding steam obeys Boyle's Law ($\mathrm{P}v$ = constant), then we have—

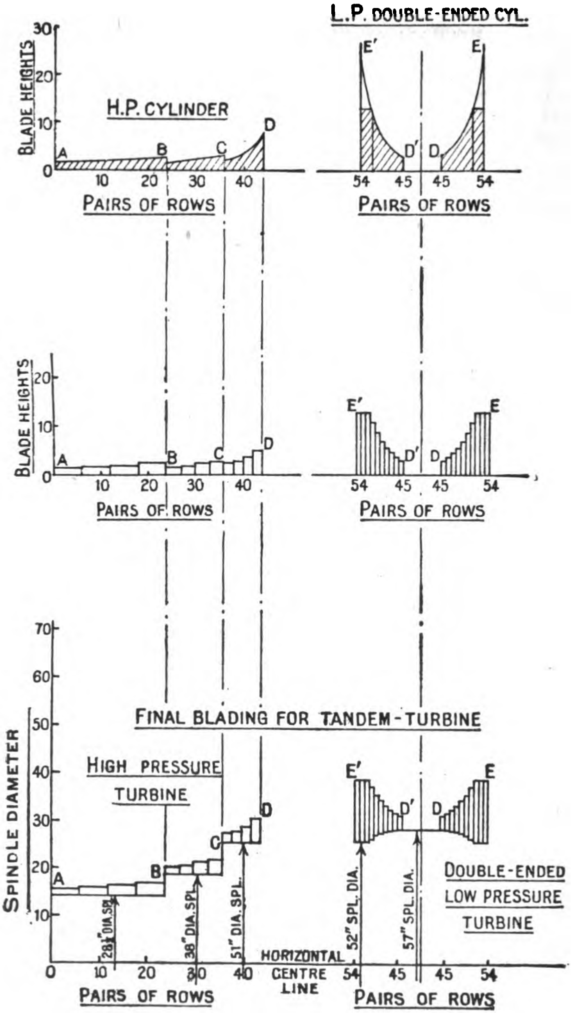
$$\mathrm{H_A} = \mathrm{P}v \log\frac{\mathrm{P_1}}{\mathrm{P}_n} = \mathrm{P}v\,(\log \mathrm{P_1} - \log \mathrm{P}_n) \qquad (31)$$

Hence

$$\frac{\mathrm{K}}{\log \mathrm{P_1} - \log \mathrm{P}_n} = \mathrm{P}v\,\frac{a^2}{a^2} \qquad (32)$$

Thus if a were made constant throughout, and PV, the "homogeneous head," were really constant throughout, the equation (34) above would be that of a straight line. That is, the total K plotted on a base of the logarithms of the successive steam pressures along the blading, would give a straight line. This is shown in Figs. 18 and 26, where the above assumptions are made.

In actual design, however, although the velocity-ratio of all the stages may be made approximately constant, the "homogeneous head" PV is by no means constant, and we arrive in actual practice at a curve somewhat as indicated by the line $\mathrm{AB_1}$. Although the divergence of this curve from the straight line AB is very considerable, yet the theory which has just been discussed affords a useful method of procedure when preliminary calculations are being made.



Figs. 23, 24, 25.

Reaction Turbine Blade-heights.

It should be noted that, by double-ending the low-pressure turbine, the total output for a given leaving loss is twice that which can be obtained from a single cylinder, in which the same maximum mean-blade speed and exhaust vacuum are adopted.

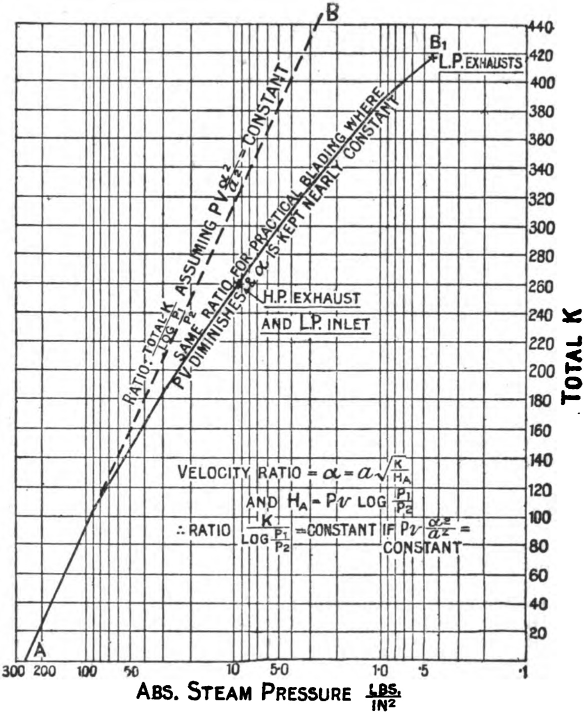


FIG. 26.—REACTION TURBINE BLADING DESIGN.

Calculation of blade-heights—

Turning back to equation (30), we have in addition the fundamental relation that $Qv = AV$, or

$$V = 144Qv/3{,}600\ A.$$

$$h = \frac{1}{3{\cdot}075 \times 10^7}\,\frac{RNQ}{adP} = \left(\frac{RQ}{3{\cdot}075 \times 10^7}\right) \times \frac{N}{adP} = \beta.\frac{n}{adP} \quad (33)$$

This is an approximate relation giving the necessary blade height for an expansion in a reaction turbine, and it is applied as follows—

Back Pressure Turbine.

$N = 3{,}000$ r.p.m. Steam at 180 lbs./ins.^{2} gauge.
Initial superheat, 150° F. Exhaust pressure 54 lbs./ins.^{2} gauge.
Steam per hour $= Q = 64{,}500$ lbs.
$H_A =$ available heat per lb. of steam $= 86$ B.th.u.

Expansion No.	N No. of Pairs.	Diameter of Spindle.	Assumed m. Dia.	$\frac{U}{V} = a$	H_A	P. abs. $\frac{\text{lbs.}}{\text{in.}^2}$	dP	$\frac{N}{a}$	$\frac{N}{a \cdot dP}$	$h = \frac{N}{a\,dP}$	K 3,000. RPM	Actual h.
1	8	$16\frac{1}{2}$″	$17\frac{15}{16}$″	0·79	28·0	174	40	10·12	0·253	1·59	23·2	$1\frac{7}{16}$″
2	8	,,	$18\frac{3}{16}$″	,,	28·5	134	35	,,	0·290	1·82	23·8	$1\frac{11}{16}$″
3	8	,,	$18\frac{9}{16}$″	,,	29·5	99	30	,,	0·338	2·14	24·8	$2\frac{1}{16}$″
			exhaust pressure 69									

$$\text{here } \beta = \frac{3{,}000 \times 64{,}500}{3{\cdot}075 \times 10^7} = 6{\cdot}30$$

The values of dP (the necessary pressure drops) are found from an assumed value of the velocity ratio a. The number of rows per expansion, the speed of the turbine, and the assumed mean diameters give the approximate "K" value.

Then we have

$$a = {\cdot}87\sqrt{K/\text{B.Th.U.}},$$

from which the B.Th.U./lb. can be found for each expansion. (*See* Fig. 27.) Starting with an assumed P_1 value on the Mollier diagram at (A), we set these heat drops down the vertical ABCD, and read off the pressures they entail. (For the purposes of preliminary calculation, it is quite sufficient to use the vertical AD, instead of the sloping stepped line AE.)

This calculation gives us the approximate values of the blade heights required, and from them we are able to form a closer idea of the actual values of the mean diameters. It will be seen that the assumed mean diameters are so close to the required diameters, that the blade heights from them (by difference from the spindle diameters) are quite accurate enough. In making the final calculation for the blade heights, the procedure is reversed, that is to say the blade heights are assumed, and the resulting velocity ratios are worked out from them. This is done because, in practice, the blade heights have to be made to the nearest sixteenth of an inch for commercial reasons.

In the case of compound Rateau steam turbines, where the pressure ratio does not exceed the critical value, the method indicated above is applicable, with a slight alteration to the velocity ratio relation—

$$a = \frac{U}{V} = a\sqrt{\frac{K}{H_A}}$$

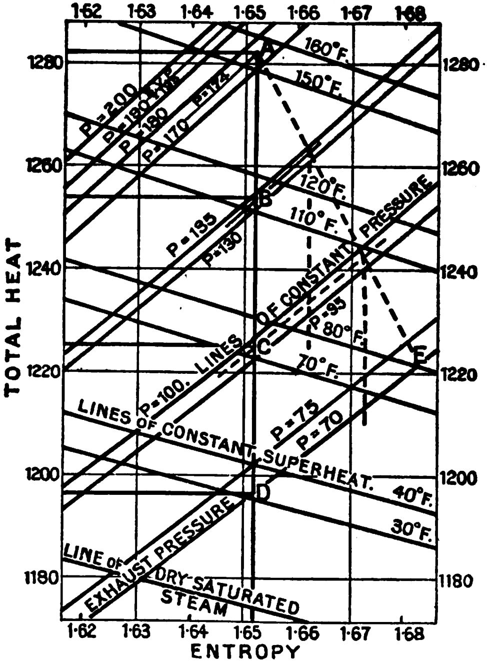


Fig. 27.—Reaction Turbine Design by Means of the "Mollier" Total-heat-entropy Diagram.

For the Rateau type of turbine (single-row velocity wheels), as the heat drop takes place in the nozzles only, the velocity of jets will be $\sqrt{2}$ higher than in the case of a "reaction pair" on the same mean diameter, and working with the same heat drop, so that the relation becomes

$$a = \frac{U}{V} = 0{\cdot}62\sqrt{\frac{K}{H_A}}$$

Different designers, of course, use different methods of calculation, and the foregoing is merely an indication of one of the best methods of tackling the problem. It will readily be realized that no strictly accurate method of calculation can be used, owing to the difficulties involved. In the first place, no strictly accurate and convenient expression can be found to represent the actual expansion law of steam. The expression

$$Pv = C$$

is not even approximately true for this working fluid. To take an example, we have—

Initial steam pressure 200 lbs./ins.^{2} gauge, initial superheat 200° F., the initial specific volume = 2·84 ft.^{3}/lb.

Hence the initial "homogeneous head" $H = 144Pv = (144 \times 215 \times 2{\cdot}84)$ ft. = 88,000.

At the exhaust of a condensing turbine, working with a 29 in. vac., say, we have final specific volume about 580ft.^{3}/lb., so that

The final "homogeneous head" $(144 \times 0{\cdot}49 \times 580) = 41{,}000$ ft. only.

Another approximate equation used to represent the expansion law is the so-called "adiabatic" law—

$$Pv^n = C\ ;$$

but here, again, the law only holds good over a short range of expansion for any one value of n. In practice, therefore, when calculations for the blading for any steam conditions have to be made in a reasonable time, and with a reasonable expenditure of effort, approximate methods must be resorted to. These requirements offer full scope to the skill and judgment of the individual designer. As mentioned before, one of the first requisites is an accurate set of steam tables, because the accuracy of the blading calculations depends primarily upon correct values of entropy, specific volume, total heat, etc.

CAPACITY OF STEAM TURBINES.

The total quantity of steam that will pass per hour through a given steam turbine, under given steam conditions, is easily determined, although the power generated by that steam is another matter.

In the case of Curtis steam turbines, and Rateau steam turbines where the critical pressure ratio is exceeded in the first compartment, the primary set of nozzles only determine the amount of steam passed; and, as already shown, the discharge is given by the relation

$$\frac{Q}{P_1} = \frac{A_t}{c} \text{ (\textit{see} equation 22)}$$

In the case of reaction turbines (and Rateau turbines where the critical pressure ratio in the first stage is not exceeded), the calculation is not quite so simple. Taking the schedule given for the back pressure reaction turbine worked out on page 550, we have—

$$\delta S = \frac{N}{h^2 D^2} = \text{a constant for each expansion} \qquad (34)$$

Writing these down for each, we get—

Expansion No.	N.	*h*.	D = Mean Diameter.	$\frac{N}{h^2D^2}$.	P abs.
1	8	$1\frac{7}{16}''$	$17\frac{15}{16}''$	0·012	174
2	8	$1\frac{1}{16}''$	$18\frac{3}{16}''$	0·0085	—
3	8	$2\frac{1}{16}''$	$19\frac{9}{16}''$	0·00544	—
					exh: 69

If the sum of the values of N/h^2D^2 for each expansion is equal to S, and then the steam capacity Q of the reaction turbine is given by the relation

$$P_1^2 - P_n^2 = \frac{7}{10^8 k^2} \cdot \frac{H}{144} \cdot Q^2 S \qquad (35)$$

In condensing turbines, the value of $(P_n)^2$ becomes negligible compared with that of the square of the initial pressure; and we may then write simply

$$\frac{Q}{P_1} = \frac{\lambda}{\sqrt{S}} \qquad (36)$$

an equation analogous to the formula (22) given above for the impulse types. Here, again, the accuracy of the result depends upon the correct experimental determination of both c and λ. We have already made some remarks (p. 528) as to the value of c, and we need here only consider the value of λ.

It can easily be shown, from first principles, that the value of λ is given (theoretically and approximately) by the relation

$$\lambda = \frac{1}{\sqrt{\frac{7}{10^8 k^2} \cdot \frac{H}{144}}} \qquad (37)$$

as indicated on page 549, equation (33), for reaction turbines; and

$$\lambda = \frac{1}{\sqrt{\frac{3 \cdot 5}{10^8 k^2} \cdot \frac{H}{144}}} \qquad (38)$$

for compound impulse turbines, where the critical pressure ratio is not exceeded in any stage.

In actual practice, dummy piston and blade clearance leakage in reaction turbines, and gland and diaphragm leakage in some Rateau turbines, increase the capacity calculated on the blading areas only.

Otherwise, however, it will be seen that (from the above relation for λ) λ depends only on k, the opening coefficient of the blades (which is accurately known for a given turbine by actual measurement), and upon H, the "homogeneous head" of the steam. This last may be described as the "effective head" on the turbine, and is equal to $\sigma 144P_1v_1$, (where σ = a coefficient). H is always lower than the initial value H_1 ($= 144P_1v_1$).

This "effective" value of H must always remain empirical, for the reason already given, namely, that no accurate and convenient expression can be found for the relation between "H" and "P" for the expansion of steam.

Were Boyle's Law ($Pv = C$) accurately true for steam, the matter would be simple, for "H" would be constant throughout the turbine.

H, then, is determined from actual tests on different turbines, designed for widely different steam conditions. We then have, for practical designs, the approximate relation

$$\frac{Q}{P_1} = \frac{78}{\sqrt{S}}$$

for high-pressure reaction condensing turbines, with ordinary radial clearance blading.

The intermediate steam pressures between P_1 and Pn in the above table can be easily obtained by a further application of equation (35), due allowance being made for the falling value of H, the homogeneous head, at each expansion.

Fig. 28 shows Parsons' "Ordinary" Radial-Clearance Reaction Blading, with thin tips. This type of blading was formerly used throughout Reaction Turbines, but is now replaced in the initial stages by a different type, described below.

In the "ordinary" blading, the tips of the blades are thinned so that if contact does take place with either cylinder or shaft, the tips will merely wear down slightly and free themselves without damage to the turbine, although the leakage area will be permanently increased. With the working clearances, however, found in practice to be necessary for safe running, the leakage of steam through the blade-clearances in the initial expansions is very serious when short blades are used in the stages which comprise those expansions, as is usually the case in all but turbines of large output. The "ordinary blading" in the high-pressure stages is, therefore, replaced by a thoroughly reliable form of adjustable axial-clearance blading, known as

PARSONS'S "END-TIGHTENED" REACTION BLADING,

in which the radial clearances are made $\frac{3}{16}''$ or more. Fig. 29 shows a perspective view of this type of blading. The spacing-pieces at the roots of the blades, instead of finishing flush with the surface of the cylinder and shaft (as is the case with "ordinary" radial clearance reaction blading) project above it and form a continuous barrier *b*, *b*.

The shrouding strip round the outer circumferences of the blade-rings is made to project over the edges of the blades on one side, in such a manner that in each "reaction pair" the shrouding strip of one row projects against the barrier of the other row. The space between the projecting edge of the shrouding strip and the corresponding barrier forms the axial clearance,

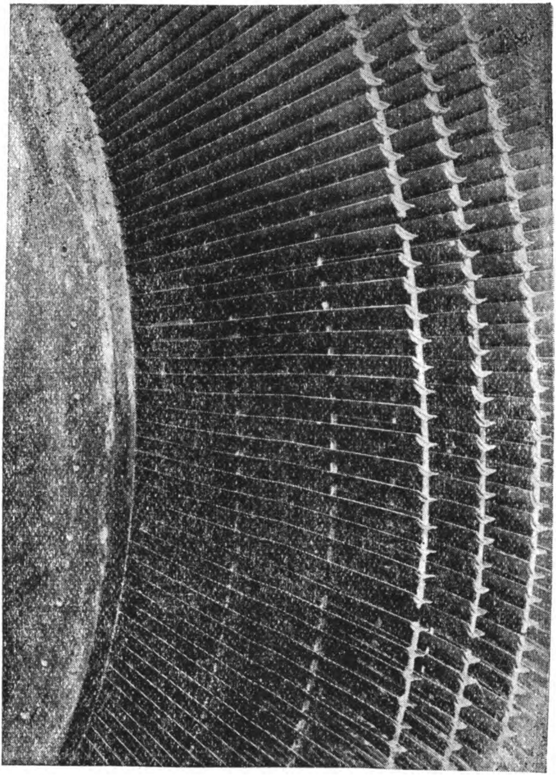


Fig. 28.—Parsons's "Ordinary" Radial Clearance Reaction Blading.

which can be adjusted to any desired amount (commonly 0·008 ins. up to 0·015 ins. in large sizes), by means of the thrust bearing at the steam end of the turbine shaft. The rigid

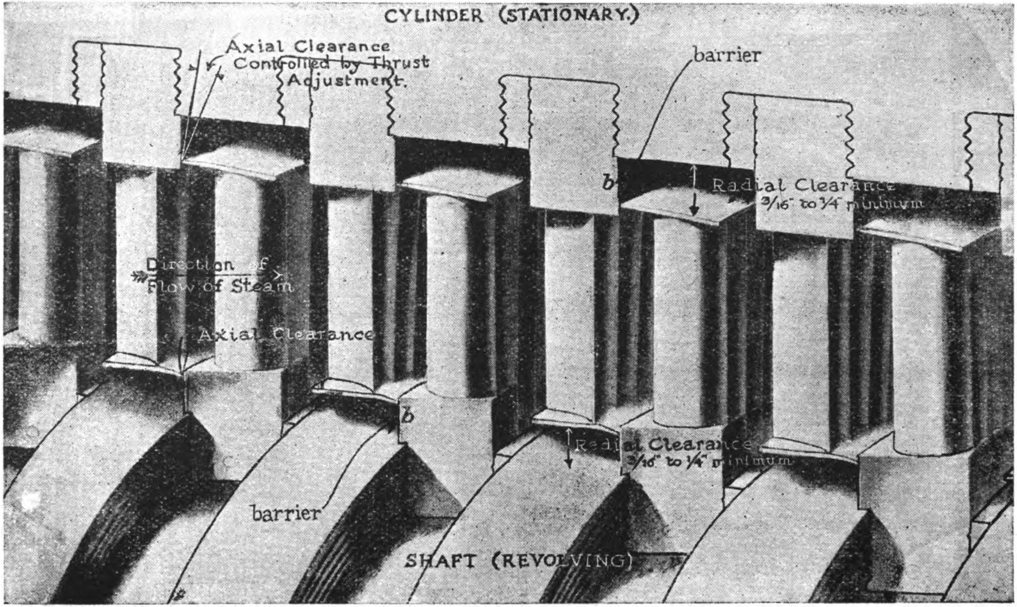


FIG. 29.—PARSONS "END-TIGHTENED" REACTION BLADING.

The accompanying table of actual official test results were obtained on the official trials of four Parsons Reaction turbines of identical design, fitted with "End-Tightened" blading.

FOUR TANDEM REACTION TURBINES AND PARSONS 3-PHASE ALTERNATORS

10,000 *K.W. Economical Load, at* 2,400 *R.P.M.*

No.	lbs. S.V.P. Gauge.	S.V.T. °F.	Supht. °F.	Exhaust Vac. (bar. 30").	Load KW.	R.P.M.	Steam Consumption lbs./KW. hr.	Remarks.
1	251	687	281	29·04"	9,991	2,442	10·04	After 30 months' running
2	234	707	306	28·96"	9,844	2,414	10·14	,, 30 ,, ,,
3	241	678	275	29·00"	10,053	2,441	10·28	,, 1 ,, ,,
4	237	696	295	29·04"	10,172	2,440	10·04	,, 1 ,, ,,

It will be noted that the steam consumptions of Nos. 1 and 2 in the above table, after 30 months' running, are almost identical with those obtained on Nos. 3 and 4 after only 1 month's running.

construction of the shaft obviates all danger of axial contact by "flutter" (*cf.* page 537).

Towards the exhaust end of a condensing turbine (where good blade-heights can be obtained), the leakage caused by the necessary radial working-clearances (in the case of the "ordinary" blading) is small—probably no greater than that which would be experienced with the "end-tightened" type, if fitted in that part of the turbine.

The present-day tendency, however, is to extend the use of the latter towards the exhaust end, on account of the advantage it possesses in permitting a radial clearance of $\frac{3}{16}$ ins. or more.

CRITICAL SPEEDS OF ROTATION OF TURBINE SHAFTS.

Suppose that, in Fig. 30 we have a vertical shaft, stationary about an axis XX, and that an intermittent (periodic) force F is applied in the direction CB, to the centre of the shaft.

The shaft will then vibrate about the axis XX, and if the force F is applied at correct intervals, the effect will be cumulative, and the amplitude of vibration will become greater and greater, with the result that the shaft will ultimately be bent beyond the elastic-limit, and may even be fractured.

Now suppose that the shaft is made to rotate about the axis XX, and that the periodic force F is replaced by a constant force F_1 acting in a fixed direction CB. The shaft will then be compelled to rotate in a bent shape, such as XAX. Consider the point *a*, first when it is on the convex side. One half of a revolution later, it will be at *b*, on the concave side of the shaft. A further half-revolution will bring it back to the convex or tension side again.

Thus, the rotation of the shaft in this bent condition is equivalent to bending a non-rotating shaft backwards and forwards in one plane, i.e. vibrating it. Let the natural frequency of vibration be *f*. Then if the number of revolutions per second is numerically equal to *f*, the alternate tension and compression of points on the shaft synchronize with the natural frequency of vibration.

The effect of the force F_1 will thus be cumulative, and the shaft will bend more and more, no matter how small F_1 may be. The shaft will now be rotating in an unstable condition, and it is said to be running at its fundamental "critical speed." Higher "critical speeds" also exist, corresponding to different modes of vibration of the shaft.

On increasing the speed of rotation above the fundamental "critical" value, the first result, however, will be a return to the stable condition, because the forced vibration of the shaft will now no longer synchronize with the natural frequency of vibration, and the shaft will "settle." A shaft may, moreover, run quite smoothly at its critical speed, if it is in good balance, and if it is not accidentally disturbed by an external force (which we have called F_1).

We have purposely premised a vertical shaft in the above discussion, in order to avoid the confusion of thought that might arise if we were considering a horizontal shaft, on account of the deflection produced by gravity. We have shown that there will be a critical speed even if the effect of gravity be eliminated (i.e. in the case of a vertical shaft).

The effect of gravity on a horizontal rotating shaft is only a

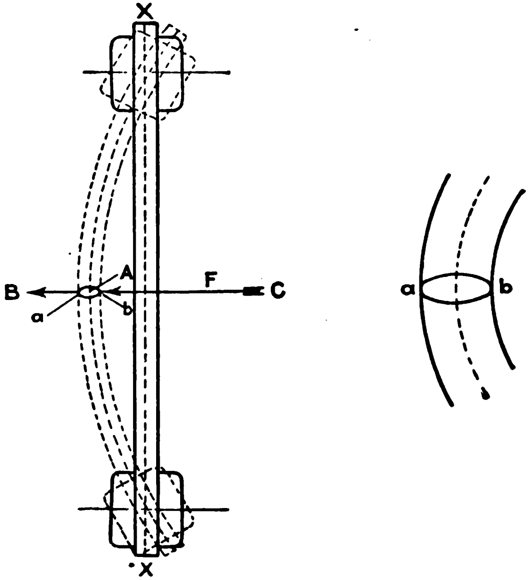


FIG. 30.—PHENOMENON OF "CRITICAL SPEED" OF A SHAFT.

special case of the above, in which gravity replaces the horizontal force F_1; but it is, however, a measure of its stiffness, and may be used to establish a relation between the critical speed and the deflection under gravity, which is of use in arriving at the approximate critical speed of a shaft. Taking the simplest mathematical case, of a vertical shaft (whose mass will be neglected, but not its flexural rigidity), carrying a concentrated mass M at the centre, then if M is set vibrating to and fro with the natural vibration frequency ω, it will execute "simple harmonic motion." If the maximum excursion from the centre line is δ_1, the force exerted by the mass M on the shaft, when δ_1 is the excursion, will be

$$F = \frac{W\omega^2\delta_1}{g}$$

But the flexural forces in the shaft itself must produce an equal and opposite force on M, which is equal to

$$P = \frac{48EI\delta_1}{L^3}$$

by the ordinary theory of flexure.

Hence, $$\frac{W\omega^2}{g} = \frac{48EI}{L^3} \text{ or } \frac{\omega^2}{g} = \frac{1}{\frac{WL^3}{48EI}} \qquad (39)$$

Now, if δ_2 is the deflection under gravity of the shaft when a weight W is placed at its centre, we have

$$\delta_2 = \frac{WL^3}{48EI} \qquad (40)$$

hence $$\frac{\omega^2}{g} = \frac{1}{\delta_2}$$

or $$\delta_2\omega^2 = g \qquad (41)$$

that is, $$n = \frac{1}{2\pi}\sqrt{\frac{g}{\delta_2}} \qquad (42)$$

giving a simple relation between the critical speed of rotation n and the deflection under gravity.

Now in the actual case of a turbine shaft, the mass M is replaced by the mass of the shaft itself (which is distributed along its length) and by that of the wheels or discs, etc., which it carries. Further, the shaft will not be of an uniform diameter throughout.

These circumstances lead to insuperable difficulties in the way of a rigid mathematical solution of the problem, but in general the approximate fundamental critical speed may be found by writing

$$\delta_2\omega^2 = \gamma g \qquad (43)$$

where γ is a coefficient which can be justified and approximately arrived at by theory, but which is best determined by actual experiment.(7)

K. Baumann, in his paper read before the Institution of Electrical Engineers in 1912 (vol. 48, page 813), says—

"... By well-known methods the critical speed and deflection due to gravity were calculated (as accurately as possible by different engineers) with the invariable result that

$$\omega^2\delta_2 = (1{\cdot}07 \text{ to } 1{\cdot}08)g \qquad (44)$$

in the case of shafts as used in impulse turbines, blowers, and compressors."

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(7) *See* list of references at end of article.

The application of this formula to reaction turbine shafts is, however, not quite so satisfactory, owing to the unsymmetrical design, and large changes in diameter. The practice of *Messrs. C. A. Parsons & Co., Ltd.*, is to make the turbine shaft wherever possible of a single forging, without built-up formation of any kind, and machined out into wheels wherever the diameter is large enough to permit it.

The limit to this type of shaft is the size of forging that can be commercially obtained, and in practice such shafts are eminently satisfactory, especially for high-speed work. The critical speed as determined by calculation is exceedingly difficult to verify in practice when this formation is adopted, and no trouble of any kind is experienced. Such shafts require very little balancing.

In this connection, the vibration which may take place when a turbine shaft is run at its critical speed, and which may be severe, must not be confused with that due to want of running "balance." If a shaft is not correctly dynamically balanced, then it will vibrate more or less at all speeds.

The importance of running balance in a turbine shaft cannot be over-estimated, as the following simple illustration will show.

Suppose that a certain part of a shaft (whose total weight is, say, 8 tons) is heavy to the extent of one pound weight at two feet radius. Then, if the speed of rotation be, say, 3,000 r.p.m., the "out of balance" centrifugal force will be

$$F = \frac{W\omega^2R}{g} = \frac{1 \times (100\pi)^2 \times 2}{32{\cdot}2} = 6{,}150 \text{ lbs.}$$

that is to say, about 6,000 times gravity, or 2/3–4 tons.

There are other higher critical speeds than the first or "fundamental" critical speed, and it is quite normal practice to run shafts well above the first and well below the second. A shaft may even run quite well actually on one of its critical speeds, and need not necessarily vibrate, but it is then in an unstable condition, a state of affairs which naturally should be avoided in practice.

STRESSES IN STEAM TURBINES.

The modern steam turbine is, in general, designed to develop the maximum possible output at the highest possible speed of rotation, the latter being usually dictated (within limits) by extraneous considerations.

Let V = the final absolute velocity (with which the steam leaves the last turbine wheel and enters the exhaust), corresponding to the allowable leaving loss (see page 524);
λ = the safety factor;
F = the ultimate strength of the material of the wheel;
D = the mean diameter of the wheel;
ω = the angular velocity of the wheel.

Then $F = \lambda\sigma\omega^2D^2$ . . . . (45)
where σ = a constant (see page 565).

Also $Qv = AV$ and $h = D/5$ (see page 524).

Then by substitution we get

$$Q = \frac{\pi kV}{5v\lambda\sigma} \cdot \left(\frac{F}{\omega^2}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad (46)$$

So that the maximum output obtainable for a given vacuum and leaving loss depends upon the ultimate strength of the material, and on the speed of rotation. (*See* equation (11).)

Now, for a given speed of rotation, the stress in a disc depends on the diameter. In order, therefore, to develop the maximum possible output at the highest possible speed, the speed is first fixed (by other considerations, such as frequency in the case of alternating current generators), and the wheel diameter can then be made just as large as will entail the maximum permissible stresses due to rotation. The maximum quantity of steam (for the given "leaving loss") that this wheel will pass can then be determined. This gives the maximum obtainable output.*

We have seen that the overall performance of a turbine depends (primarily) on the overall "velocity-ratio" at which the several compartments or stages are operating, the said overall velocity-ratio being given by the equation

$$a = a\sqrt{\frac{K}{H_A}}$$

where K = (mean-diameter × R.P.M.)^2 × N × 10^{-9} for the whole turbine.

The expression for K shows that the necessary combination of wheels and blade rings can be most easily obtained by using large diameters, and high speed of rotation, rather than by large number of compartments, because K depends on the square of the mean diameter and of the revolutions, but only on the number of compartments direct.

For modern high initial steam pressures and temperatures and high exhaust vacuum, it is by no means easy to obtain the requisite K when the output required is small (about 1,000 k.w.), and hence the value of mechanical gearing, which enables the speed of the turbine to be increased (thus increasing K) without the necessity for making a very long, flexible shaft, or the diameters too large for the output required.

In modern marine installations, where reduction gearing is employed, the propeller may be made to rotate at the speed most suitable for efficiency, whilst the turbines may be run up to any suitable speed that is consistent with safety.

In the case of direct-current dynamos, the best modern practice now centres round slow-speed machines, so that if a turbine is to be used to drive them, it is necessary to interpose reduction gearing.

For many years, however, before the perfection of such gearing, fairly satisfactory high-speed turbo-generators were constructed, the turbine being direct coupled to the dynamo, but a limitation is imposed on this procedure owing to the fact that turbines and dynamos do not go up in size together.

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* *See* K. Baumann, *Journal I.E.E.* vol. 48, page 806.

If the dimensions of a turbine be doubled all over, and it be run at one half of the speed in order to maintain the same stresses, then it will have four times the steam capacity, and will in consequence develop four times the output. A dynamo, on the other hand, goes up in output almost inversely as the speed only, so that in the case of a fairly large output of, say, 3,000 k.w., the proper speed of rotation for it will be such that a turbine, if running direct coupled, would be capable of giving an output far in excess of that of the dynamo. If the turbine were cut down in size so that its output suited that of the dynamo (3,000 k.w.) then it would be working at a poor overall velocity-ratio owing to the low surface speeds, and the economy also would be poor.

Hence in modern designs, gearing is interposed, so that the turbine may be run up to a higher speed, a much smaller frame utilized, and yet the requisite K obtained in order to give good economy. Further, the dynamo design being now an ordinary slow-speed one, ordinary materials and construction may be used, which for moderate outputs are much less costly than a pure high-speed design, although the latter may be smaller in size.

In the case of alternators, these go up in output under practically the same law as in the case of turbines, that is to say inversely as the square of the speed of rotation. On the other hand, definite standard frequencies of alternation of potential are required. In Great Britain many frequencies at present are unfortunately in use, ranging from 25 complete cycles per second up to 100, the tendency being to adopt 50 cycles as the standard. If the alternator has p pairs of poles, and is driven at N revolutions per minute, then the frequency of alternation will be—

$$f = Np/60 \text{ cycles per second} \qquad (47)$$

Thus, if we adopt 50 cycles per second, then we have the following speeds of rotation, and these only—

Frequency f.	No. of pairs of poles, p.	R.p.m. N.
50	1	3,000
,,	2	1,500
,,	3	1,000
,,	4	750

Alternators of small or moderate output may, however, be geared down from the turbine speed, in order that ordinary materials and slow-speed construction may be adopted, with a view to cheapness.

STANDARD GEAR BOXES AND GEAR REDUCTION.

To facilitate commercial production, the gear boxes for standard designs are usually fixed. Thus a standard Parsons 1,000 k.w. turbo-geared generator runs at 5,000/500 r.p.m., whilst in the case of their standard 6,000 k.w. direct current design two-tandem generators on the same shaft line, each developing 3,000 k.w., are coupled to the slow-speed gear wheel, the latter being driven by high and low-pressure turbines, one driving a pinion on each side of the wheel. The expansion of steam is divided up so that each turbine develops about equal power at full load, whilst both run at 3,000 r.p.m., the dynamo speed being 300 r.p.m. A very economical and sound mechanical design is obtained in this manner.

As, however, a small alteration in pinion diameter makes a large alteration in the speed reduction ratio, it is possible to widely vary the possible gear ratio in a given gear box, without important alteration to the pattern, so that the odd speeds required for certain classes of turbo-machinery can be obtained with little alteration to standard practice (as dictated by electrical work).

RANGE OF TURBINE CYLINDERS FOR STANDARD OUTPUTS.

To enable us to design a suitable range of turbine cylinders in order to give the output required at the maximum speed, it will clearly be necessary to obtain the value and distribution of the stresses produced in discs, wheels, etc., when they are rotated, since the highest stresses in turbines occur in these parts.

Commencing with the simplest possible case—that of a rotating thin ring, we have pure hoop-tension only in it, and this is given by the equation—

$$f_t = \frac{wV^2}{2240g} \text{ lbs. per sq. inch} \tag{48}$$

$$= \left(\frac{w\,\omega^2}{2240g}\right)R^2 \text{ since } V = \omega R \tag{49}$$

where w = the weight of material per in. section, per foot run in lbs., and V is the velocity of a point on the ring in feet per second. Thus, for mild steel,

$$w = 12 \times 0{\cdot}285, \text{ and}$$

$$f_t = \frac{V^2}{21{,}000} \frac{\text{tons}}{\text{in}^2}$$

The next step is to consider a revolving circular plate, with a hole in the centre, that is, a thickened up ring. The fundamental equations, based on the elementary theory of elasticity, are

$$\frac{u}{x} = \frac{1}{E}\left(p_y - \frac{p_x}{m}\right) \text{ and } \frac{du}{dx} = \frac{1}{E}\left(p_x - \frac{p_y}{m}\right) \qquad \begin{matrix}(50)\\(51)\end{matrix}$$

where u = the radial displacement of a point at any radius x, and 1/m = Poisson's ratio, whilst p_x and p_y are the radial and circumferential stresses respectively, at any point.

This leads, after a considerable amount of mathematics, to the equations

$$P_x = \frac{w\omega^2 R_1^2}{g}\left[\frac{3+m}{8}\left\{1 + \frac{R_2^2}{R_1^2} - \frac{R_2^2}{R^2} - \frac{R^2}{R_1^2}\right\}\right] \qquad (52)$$

and

$$P_y = \frac{w\omega^2 R_1^2}{g}\left[\frac{3+m}{8}\left\{1 + \frac{R_2^2}{R_1^2} + \frac{R_2^2}{R^2}\right\} - \left(\frac{1+3m}{8}\right)\frac{R^2}{R_1^2}\right] \qquad (53)$$

Now, $w\omega^2 R_1^2/g$ represents the hoop tension in a thin rotating ring, so that the radial and hoop stress at any radius R in a rotating disc of outer radius R_1 and inner radius R_2 may be conveniently expressed in terms of the hoop stress in a thin ring of the same material, running at the same speed of rotation, and of radius R_1. The results may be graphically shown to advantage, as in Fig. 31, plotted on a base of disc radius, reckoning the outer radius as unity, and the inner radius as a fraction of it.

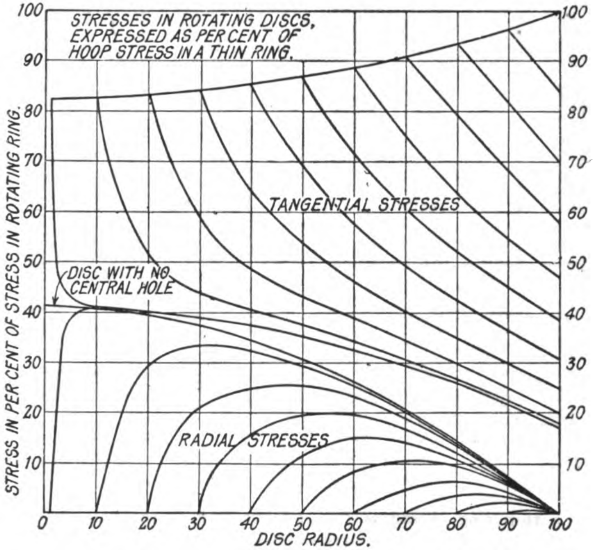

FIG. 31.

It will be noted that the hoop tension is always a maximum at the periphery of the hole, but less than that in the corresponding thin ring of radius R_1. Thus, in a rotating disc, if $R/R_1 = 0{\cdot}40$, then from the curves of hoop tension, at the periphery of the hole the hoop-tension is 86 per cent of that in a thin ring of radius R_1, and rotating at the same speed. The radial tension is a maximum at a radius $R = \sqrt{R_1 R_2}$, that is to say, at the radius which is the geometric mean of R_1 and R_2.

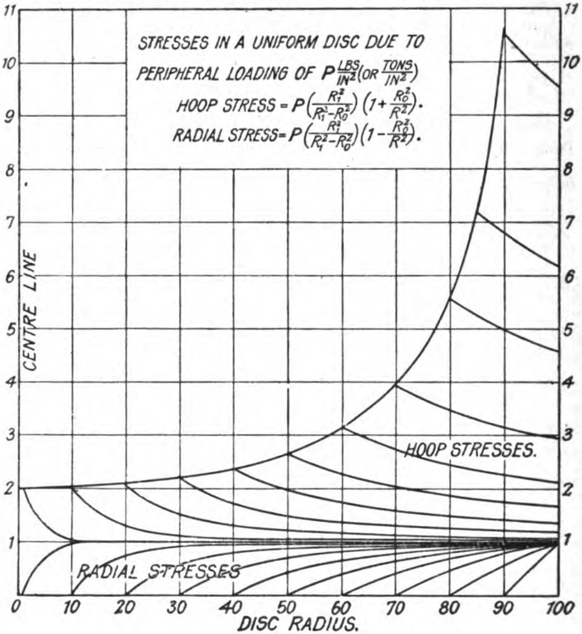

FIG. 32.

When there is no central hole, the hoop tension and radial tension are equal at the centre, and only 42 per cent of that hoop tension in a thin ring. A very small central hole, however, immediately doubles the hoop tension at the centre, whilst the radial tension there becomes zero.

The Additional Effect of peripheral loading of a disc, due to the presence of a rotating ring of blading, etc., can be approximately obtained from the ordinary theory of thick cylinders subjected to external tension, and a chart prepared (*see* Fig. 32) similar to that given in the case of plain rotating discs.

The results are then additive.

The effect of hub shrink fits on the shaft has been very effectively worked out by Professor Morley in his paper published in *Engineering* ([8]). When the discs are not simple flat discs, but have a definite profile, the mathematical theory becomes considerably more complicated. The fundamental equations are due to Professor Stodola, and are given in his treatise on "Steam Turbines."

In the design of rotating discs ([9]), it should always be borne in mind that the theory is only approximate, as in order to simplify the complication of the mathematics, we assume that the thickness of the disc is very small compared with its diameter. That is to say, we deal with a thin flat plate. This enables us to say that the displacement of any point must be radial, and that the radial and hoop tension at that point are the principal stresses and the only stresses, the axial principal stress being zero. This applies to the theory of profiled discs as well as to plain flat discs.

An ample safety factor of, say, three or four (reckoned on the ultimate strength of the material as established by careful tests) should be adopted, so that at the maximum speed at which the turbine will run (usually 10 per cent above the normal speed) there is always a reasonable and fair margin of safety.

NOTES ON OPERATION OF STEAM TURBINES.

Care and Maintenance. Oiling system (*see* diagram, Fig. 33). The most important characteristic for oil for use in turbine bearings is that it should separate freely from water, and not form an emulsion with it.

Only a pure mineral oil should be used, free from animal or vegetable fats, and it should be of a non-foaming character.

Water is the main cause of deterioration of a turbine oil, so that every effort should be made to keep it out of the oiling system, and daily inspection should be made to see that this is so. The oil that will meet the water test can be used indefinitely in a turbine by being added to from time to time, after filtering. Periodic cleaning of the oil tank and oil system is necessary, and when the oil is withdrawn for this purpose it should be filtered.

Starting up. In modern practice, it is usual not to do any preliminary heating up with the turbine at a standstill, since unequal heating, causing distortion, is likely to result. Hence the turbine is got running (slowly) as quickly as possible, so that equal heating up may be secured.

If the machine be started up condensing, the vacuum should

([8]) *See* list of references at end of article.
([9]) *See* list of references at end of article.

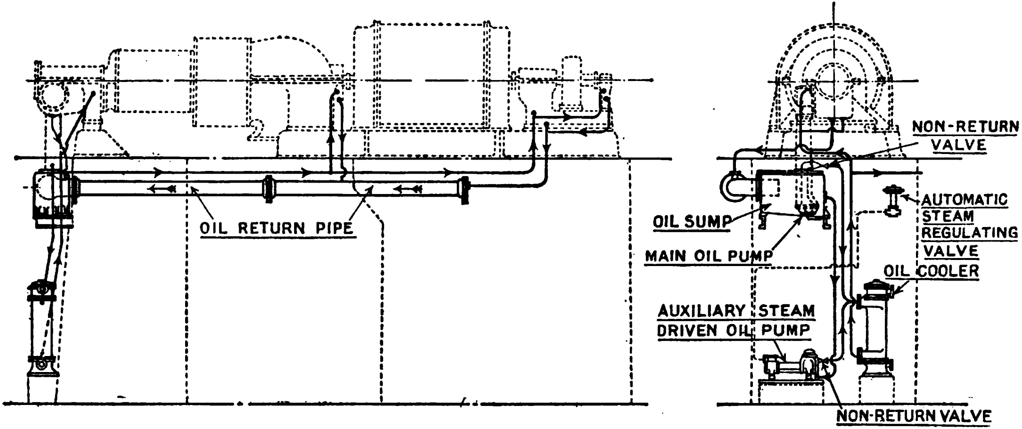

Fig. 33.—Oiling System of Typical Turbine Installation.

not exceed 25 ins. of mercury, as long as there is no load on the turbine, otherwise it may run away and trip the emergency governor, because it is very difficult to keep the governor throttle valve from passing too much steam when nearly closed, especially if the valve and seats are worn with use.

Running. When running, the temperature of the oil leaving the bearings should never exceed 170 deg. F., and it should be kept at about 140 deg. F., if possible, by means of the oil-cooler. Oil will lubricate best when the temperature of the oil from the bearings is between 110 and 140 deg. F.

Shutting Down. Having removed the load and tripped the emergency governor, the turbine stop valve should be closed and every effort made to dry out the interior of the turbine before breaking the vacuum. Corrosion of cylinders and blading is mainly caused by a damp atmosphere and water remaining inside the machine when it is lying idle. The stop valve should be properly steam tight, so as to prevent water having access to the turbine whilst not in use.

Checking of Bearings and Clearances. The bearings, which are usually split white-metal lined, should be periodically examined for possible wear down, by placing a bridge gauge over the journals, and keeping a record of the readings. The axial adjustment of the thrust block should also be periodically checked in order to make certain of the position of the shaft inside the cylinder. Constant attention of this kind reveals defects (if any) before they become serious, and the general operation of the plant is much more satisfactory than is the case if it is run until excessive wear from a preventable cause renders a complete shut-down necessary at an inopportune moment.

OTHER COMMERCIAL APPLICATIONS OF THE STEAM TURBINE.

The application of steam turbines to marine propulsion is probably better known to engineers and the community at large than any other, but there are other fields in which the turbine has a special use. In addition to pure high-pressure steam turbines, the following other types have been developed to meet the requirements of various industrial undertakings—

(1) *Low-pressure or Exhaust Steam Turbines.* These were originally developed in marine work, to take steam from a high-pressure turbine (at about atmospheric pressure), or from a reciprocating engine. The low-pressure turbine would be arranged to drive a separate propeller.

For land work, "exhaust" steam turbines have been largely adopted in collieries to utilize steam at atmospheric pressure (exhausted from piston engines) which formerly was running to waste. They are usually coupled to electric generators (either alternating or direct current), and supply the works with power. The total output from the fuel burned is approximately doubled by this means.

Pure "exhaust" steam turbines have now little or no market for land purposes, because the continuous supply of exhaust steam that was derived from reciprocating engines is gradually becoming unavailable owing to the replacement of the engines by high-pressure condensing steam turbines. The remaining source of exhaust steam is chiefly from intermittently running winding engines, which has led to the development of

(2) *Mixed-pressure Steam Turbines.* In this application of the turbine, there are separate high and low-pressure steam chests, so arranged that the governor always gives preference to low-pressure steam as long as any is available, and also automatically admits high-pressure steam to the high-pressure blading of the turbine when the low pressure supply fails, or is insufficient to meet the load on the turbine.

The demand for this type also is gradually disappearing, owing to the fact that steam-driven winding engines are coming to be replaced by electric motors supplied with power from high-pressure condensing turbines, so that even an intermittent supply of low-pressure steam would no longer be available.

(3) *Back-pressure or Reducing Turbines.* In many industrial undertakings, such as paper mills, breweries, etc., a continuous supply of low or moderate pressure steam is required for process work. Instead of installing boilers to generate this steam at the pressure at which it is required, which would be wasteful, or of passing high-pressure steam through a reducing valve, which would also be wasteful, it is much more economical to utilize high-pressure steam in a suitably designed steam turbine first, thus obtaining power for driving plant in the works, and then to exhaust it (at the requisite pressure) into the process-steam supply pipe.

Turbines which economically reduce the steam pressure in this way are known as "reducing turbines."

(4) *Pass-out or Bleeder Turbines.* A refinement of the last described design is found in the development of "pass-out" or "bleeder" turbines. Steam for process purposes may be required at two distinct pressures, and in varying amounts of each. The turbine is, therefore, arranged so that at the respective points along the blading where the pressures are appropriate, all the steam there is led out into a "pass out" chest, which is really nothing more than an automatic relief valve. If the process steam is required in less quantity than that which is entering the chest, the double beat valve in the latter automatically opens and allows the surplus steam to flow back into the turbine, there to be usefully expanded in the remainder of the blading. To ensure that all the steam shall leave the turbine and enter the pass-off chest, a short dummy piston is placed immediately after the pass-off point, so that the steam must enter the chest. This enables the pressure at the point of pass-out to be maintained under all conditions of load on the turbine.

Commonly, two such automatic chests are fitted to one turbine, each delivering steam for process purposes at a (different) appropriate pressure. The turbine may be worked as a high-pressure condensing machine if no steam is wanted for heating, or as a back-pressure machine if all the steam is required for heating (10).

It will be clear that the total amount of heating steam can fluctuate between these limits.

The parallel-flow Reaction steam turbine lends itself particularly well to "pass-out" designs of this kind, as its comparatively greater length and larger number of stages than the Impulse turbine give greater freedom of mechanical design of the plant.

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[10] *See* list of references at end of article.

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LIST OF SYMBOLS USED IN STEAM TURBINE SECTION

(WITH REFERENCE TO CONTEXT)

a = coefficient
α = velocity-ratio
A = nozzle area
b = coefficient
β = ,,
c = ,,
C = ,, in $Pv = C$ or $Pv^n = C$
δ = deflection
D = mean diameter
e = stage efficiency
ε = overall efficiency
η = diagram efficiency
E = Young's modulus
f = frequency
g = gravity
h = blade-height or length
h_a = available heat per stage
H_A = total available heat
H = static head
= "homogeneous head"
= 144 Pv feet
I = moment of Inertia
J = mechanical equivalent of heat
k = blade or nozzle opening coefficient
K = the Parsons' coefficient
L = span of shaft
l = length of nozzle arc
λ = coefficient
= safety factor

$\frac{1}{m}$ = Poissons' ratio
N = No. of compartments in a pressure compounded turbine
= No. of moving rows in a reaction turbine
P = steam pressure
= force
p = principal stress
= No. of pairs of poles
q = dryness fraction
Q = Quantity of steam discharged = W
R = Re-heat factor
= Radius of disc
= Revolutions of turbine
ρ = density of fluid
S = sum of
σ = coefficient
T = absolute temperature
U = mean blade speed
v = specific volume
V = steam velocity
$\overline{V}$ = change in velocity
W = Weight of steam discharged = Q
w = density of material
X = expansion-ratio = $\frac{P_1}{P_2}$
ω = angular velocity

TABLE A.—HYPERBOLIC LOGARITHMS

No.	Logarithm	No.	Logarithm	No.	Logarithm	No.	Logarithm
1·01	·009950	1·43	·357674	1·85	·615185	2·27	·819779
1·02	·019802	1·44	·364643	1·86	·620576	2·28	·824175
1·03	·029558	1·45	·371563	1·87	·625938	2·29	·828551
1·04	·039220	1·46	·378436	1·88	·631271	2·30	·832909
1·05	·048790	1·47	·385262	1·89	·636576	2·31	·837247
1·06	·058269	1·48	·392042	1·90	·641853	2·32	·841567
1·07	·067658	1·49	·398776	1·91	·647103	2·33	·845868
1·08	·076961	1·50	·405465	1·92	·652325	2·34	·850150
1·09	·086177	1·51	·412109	1·93	·657520	2·35	·854415
1·10	·095310	1·52	·418710	1·94	·662687	2·36	·858661
1·11	·104360	1·53	·425267	1·95	·667829	2·37	·862889
1·12	·113328	1·54	·431782	1·96	·672944	2·38	·867100
1·13	·122217	1·55	·438254	1·97	·678033	2·39	·871293
1·14	·131028	1·56	·444685	1·98	·683096	2·40	·875468
1·15	·139761	1·57	·451075	1·99	·688134	2·41	·879626
1·16	·148420	1·58	·457424	2·00	·693147	2·42	·883767
1·17	·157003	1·59	·463733	2·01	·698134	2·43	·887891
1·18	·165514	1·60	·470003	2·02	·703097	2·44	·891998
1·19	·173953	1·61	·476234	2·03	·708035	2·45	·896088
1·20	·182321	1·62	·482426	2·04	·712949	2·46	·900161
1·21	·190620	1·63	·488580	2·05	·717839	2·47	·904218
1·22	·198850	1·64	·494696	2·06	·722705	2·48	·908258
1·23	·207014	1·65	·500775	2·07	·727548	2·49	·912282
1·24	·215111	1·66	·506817	2·08	·732367	2·50	·916290
1·25	·223143	1·67	·512823	2·09	·737164	2·51	·920282
1·26	·231111	1·68	·518793	2·10	·741937	2·52	·924258
1·27	·239016	1·69	·524728	2·11	·746688	2·53	·928219
1·28	·246860	1·70	·530628	2·12	·751416	2·54	·932164
1·29	·254642	1·71	·536493	2·13	·756121	2·55	·936093
1·30	·262364	1·72	·542324	2·14	·760805	2·56	·940007
1·31	·270027	1·73	·548121	2·15	·765467	2·57	·943905
1·32	·277631	1·74	·553885	2·16	·770108	2·58	·947789
1·33	·285178	1·75	·559615	2·17	·774727	2·59	·951657
1·34	·292669	1·76	·565313	2·18	·779324	2·60	·955511
1·35	·300104	1·77	·570979	2·19	·783901	2·61	·959350
1·36	·307484	1·78	·576613	2·20	·788457	2·62	·963174
1·37	·314810	1·79	·582215	2·21	·792992	2·63	·966983
1·38	·322083	1·80	·587786	2·22	·797507	2·64	·970778
1·39	·329303	1·81	·593326	2·23	·802001	2·65	·974559
1·40	·336472	1·82	·598836	2·24	·806475	2·66	·978326
1·41	·343589	1·83	·604315	2·25	·810930	2·67	·982078
1·42	·350656	1·84	·609765	2·26	·815364	2·68	·985816

TABLE A. (*Continued*)

No.	Logarithm	No.	Logarithm	No.	Logarithm	No.	Logarithm
2·69	·989541	3·11	1·13462	3·53	1·26129	3·95	1·37371
2·70	·993251	3·12	1·13783	3·54	1·26412	3·96	1·37624
2·71	·996948	3·13	1·14103	3·55	1·26694	3·97	1·37876
2·72	1·00063	3·14	1·14422	3·56	1·26976	3·98	1·38128
2·73	1·00430	3·15	1·14740	3·57	1·27256	3·99	1·38379
2·74	1·00795	3·16	1·15057	3·58	1·27536	4·00	1·38629
2·75	1·01160	3·17	1·15373	3·59	1·27815	4·01	1·38879
2·76	1·01523	3·18	1·15688	3·60	1·28093	4·02	1·39128
2·77	1·01884	3·19	1·16002	3·61	1·28370	4·03	1·39377
2·78	1·02245	3·20	1·16315	3·62	1·28647	4·04	1·39624
2·79	1·02604	3·21	1·16627	3·63	1·28923	4·05	1·39871
2·80	1·02961	3·22	1·16938	3·64	1·29198	4·06	1·40118
2·81	1·03318	3·23	1·17248	3·65	1·29472	4·07	1·40364
2·82	1·03673	3·24	1·17557	3·66	1·29746	4·08	1·40609
2·83	1·04027	3·25	1·17865	3·67	1·30019	4·09	1·40854
2·84	1·04380	3·26	1·18172	3·68	1·30291	4·10	1·41098
2·85	1·04731	3·27	1·18478	3·69	1·30562	4·11	1·41342
2·86	1·05082	3·28	1·18784	3·70	1·30833	4·12	1·41585
2·87	1·05431	3·29	1·19088	3·71	1·31103	4·13	1·41827
2·88	1·05779	3·30	1·19392	3·72	1·31372	4·14	1·42069
2·89	1·06125	3·31	1·19694	3·73	1·31640	4·15	1·42310
2·90	1·06471	3·32	1·19996	3·74	1·31908	4·16	1·42551
2·91	1·06815	3·33	1·20297	3·75	1·32175	4·17	1·42791
2·92	1·07158	3·34	1·20597	3·76	1·32441	4·18	1·43031
2·93	1·07500	3·35	1·20896	3·77	1·32707	4·19	1·43270
2·94	1·07840	3·36	1·21194	3·78	1·32972	4·20	1·43508
2·95	1·08180	3·37	1·21491	3·79	1·33236	4·21	1·43746
2·96	1·08518	3·38	1·21787	3·80	1·33500	4·22	1·43983
2·97	1·08856	3·39	1·22082	3·81	1·33762	4·23	1·44220
2·98	1·09192	3·40	1·22377	3·82	1·34025	4·24	1·44456
2·99	1·09527	3·41	1·22671	3·83	1·34286	4·25	1·44691
3·00	1·09861	3·42	1·22964	3·84	1·34547	4·26	1·44926
3·01	1·10194	3·43	1·23256	3·85	1·34807	4·27	1·45161
3·02	1·10525	3·44	1·23547	3·86	1·35066	4·28	1·45395
3·03	1·10856	3·45	1·23837	3·87	1·35325	4·29	1·45628
3·04	1·11185	3·46	1·24126	3·88	1·35583	4·30	1·45861
3·05	1·11514	3·47	1·24415	3·89	1·35840	4·31	1·46093
3·06	1·11841	3·48	1·24703	3·90	1·36097	4·32	1·46325
3·07	1·12167	3·49	1·24990	3·91	1·36353	4·33	1·46556
3·08	1·12492	3·50	1·25276	3·92	1·36609	4·34	1·46787
3·09	1·12817	3·51	1·25561	3·93	1·36863	4·35	1·47017
3·10	1·13140	3·52	1·25846	3·94	1·37118	4·36	1·47247

TABLE A. (*Continued*)

No.	Logarithm	No.	Logarithm	No.	Logarithm	No.	Logarithm
4·37	1·47476	4·79	1·56653	5·21	1·65057	5·63	1·72810
4·38	1·47704	4·80	1·56861	5·22	1·65249	5·64	1·72988
4·39	1·47932	4·81	1·57069	5·23	1·65441	5·65	1·73165
4·40	1·481604	4·82	1·57277	5·24	1·65632	5·66	1·73342
4·41	1·48387	4·83	1·57484	5·25	1·65822	5·67	1·73518
4·42	1·48613	4·84	1·57691	5·26	1·66013	5·68	1·73695
4·43	1·48839	4·85	1·57897	5·27	1·66203	5·69	1·73871
4·44	1·49065	4·86	1·58103	5·28	1·66392	5·70	1·74046
4·45	1·49290	4·87	1·58309	5·29	1·66581	5·71	1·74221
4·46	1·49514	4·88	1·58514	5·30	1·66770	5·72	1·74396
4·47	1·49738	4·89	1·58719	5·31	1·66959	5·73	1·74571
4·48	1·49962	4·90	1·58923	5·32	1·67147	5·74	1·74745
4·49	1·50185	4·91	1·59127	5·33	1·67335	5·75	1·74919
4·50	1·50407	4·92	1·59330	5·34	1·67522	5·76	1·75093
4·51	1·50629	4·93	1·59533	5·35	1·67709	5·77	1·75267
4·52	1·50851	4·94	1·59736	5·36	1·67896	5·78	1·75440
4·53	1·51072	4·95	1·59938	5·37	1·68082	5·79	1·75613
4·54	1·51292	4·96	1·60140	5·38	1·68268	5·80	1·75785
4·55	1·51512	4·97	1·60341	5·39	1·68454	5·81	1·75958
4·56	1·51732	4·98	1·60542	5·40	1·68639	5·82	1·76130
4·57	1·51951	4·99	1·60743	5·41	1·68824	5·83	1·76301
4·58	1·52169	5·00	1·60943	5·42	1·69009	5·84	1·76473
4·59	1·52388	5·01	1·61143	5·43	1·69193	5·85	1·76644
4·60	1·52605	5·02	1·61343	5·44	1·69377	5·86	1·76814
4·61	1·52822	5·03	1·61542	5·45	1·69561	5·87	1·76985
4·62	1·53039	5·04	1·61740	5·46	1·69744	5·88	1·77155
4·63	1·53255	5·05	1·61938	5·47	1·69927	5·89	1·77325
4·64	1·53471	5·06	1·62136	5·48	1·70110	5·90	1·77495
4·65	1·53686	5·07	1·62334	5·49	1·70292	5·91	1·77684
4·66	1·53901	5·08	1·62531	5·50	1·70474	5·92	1·77833
4·67	1·54115	5·09	1·62727	5·51	1·70656	5·93	1·78002
4·68	1·54329	5·10	1·62924	5·52	1·70837	5·94	1·78170
4·69	1·54543	5·11	1·63119	5·53	1·71018	5·95	1·78339
4·70	1·54756	5·12	1·63315	5·54	1·71199	5·96	1·78507
4·71	1·54968	5·13	1·63510	5·55	1·71379	5·97	1·78674
4·72	1·55180	5·14	1·63705	5·56	1·71559	5·98	1·78842
4·73	1·55392	5·15	1·63899	5·57	1·71739	5·99	1·79009
4·74	1·55663	5·16	1·64093	5·58	1·71918	6·00	1·79175
4·75	1·55814	5·17	1·64287	5·59	1·72097	6·01	1·79342
4·76	1·56024	5·18	1·64480	5·60	1·72276	6·02	1·79508
4·77	1·56234	5·19	1·64673	5·61	1·72455	6·03	1·79674
4·78	1·56444	5·20	1·64865	5·62	1·72633	6·04	1·79804

TABLE A. (*Continued*)

No.	Logarithm	No.	Logarithm	No.	Logarithm	No.	Logarithm
6·05	1·80005	6·47	1·86717	6·89	1·93007	7·31	1·98924
6·06	1·80170	6·48	1·86872	6·90	1·93152	7·32	1·99061
6·07	1·80335	6·49	1·87026	6·91	1·93296	7·33	1·99197
6·08	1·80500	6·50	1·87180	6·92	1·93441	7·34	1·99333
6·09	1·80664	6·51	1·87333	6·93	1·93585	7·35	1·99470
6·10	1·80828	6·52	1·87487	6·94	1·93730	7·36	1·99605
6·11	1·80992	6·53	1·87640	6·95	1·93874	7·37	1·99741
6·12	1·81156	6·54	1·87793	6·96	1·94017	7·38	1·99877
6·13	1·81319	6·55	1·87946	6·97	1·94161	7·39	2·00012
6·14	1·81482	6·56	1·88099	6·98	1·94304	7·40	2·00148
6·15	1·81645	6·57	1·88251	6·99	1·94448	7·41	2·00283
6·16	1·81807	6·58	1·88403	7·00	1·94591	7·42	2·00417
6·17	1·81969	6·59	1·88555	7·01	1·94733	7·43	2·00552
6·18	1·82131	6·60	1·88706	7·02	1·94876	7·44	2·00687
6·19	1·82293	6·61	1·88858	7·03	1·95018	7·45	2·00821
6·20	1·82454	6·62	1·89009	7·04	1·95160	7·46	2·00955
6·21	1·82616	6·63	1·89160	7·05	1·95302	7·47	2·01089
6·22	1·82776	6·64	1·89311	7·06	1·95444	7·48	2·01223
6·23	1·82937	6·65	1·89461	7·07	1·95586	7·49	2·01356
6·24	1·83098	6·66	1·89611	7·08	1·95727	7·50	2·01490
6·25	1·83258	6·67	1·89761	7·09	1·95868	7·51	2·01623
6·26	1·83418	6·68	1·89911	7·10	1·96009	7·52	2·01756
6·27	1·83577	6·69	1·90061	7·11	1·96150	7·53	2·01889
6·28	1·83736	6·70	1·90210	7·12	1·96290	7·54	2·02022
6·29	1·83896	6·71	1·90359	7·13	1·96431	7·55	2·02154
6·30	1·84054	6·72	1·90508	7·14	1·96571	7·56	2·02287
6·31	1·84213	6·73	1·90657	7·15	1·96711	7·57	2·02419
6·32	1·84371	6·74	1·90806	7·16	1·96850	7·58	2·02551
6·33	1·84531	6·75	1·90954	7·17	1·96990	7·59	2·02683
6·34	1·84687	6·76	1·91102	7·18	1·97129	7·60	2·02814
6·35	1·84845	6·77	1·91250	7·19	1·97269	7·61	2·02946
6·36	1·85002	6·78	1·91397	7·20	1·97408	7·62	2·03077
6·37	1·85159	6·79	1·91545	7·21	1·97546	7·63	2·03208
6·38	1·85316	6·80	1·91692	7·22	1·97685	7·64	2·03339
6·39	1·85473	6·81	1·91839	7·23	1·97823	7·65	2·03470
6·40	1·85629	6·82	1·91985	7·24	1·97962	7·66	2·03601
6·41	1·85785	6·83	1·92132	7·25	1·98100	7·67	2·03731
6·42	1·85941	6·84	1·92278	7·26	1·98237	7·68	2·03861
6·43	1·86097	6·85	1·92424	7·27	1·98375	7·69	2·03992
6·44	1·86252	6·86	1·92570	7·28	1·98513	7·70	2·04122
6·45	1·86408	6·87	1·92716	7·29	1·98650	7·71	2·04251
6·46	1·86562	6·88	1·92861	7·30	1·98787	7·72	2·04381

TABLE A. (*Continued*)

No.	Logarithm	No.	Logarithm	No.	Logarithm	No.	Logarithm
7·73	2·04510	8·15	2·09801	8·57	2·14826	8·99	2·19611
7·74	2·04640	8·16	2·09924	8·58	2·14943	9·00	2·19722
7·75	2·04769	8·17	2·10046	8·59	2·15059	9·01	2·1983
7·76	2·04898	8·18	2·10169	8·60	2·15176	9·02	2·1994
7·77	2·05027	8·19	2·10291	8·61	2·15292	9·03	2·2006
7·78	2·05155	8·20	2·10413	8·62	2·15408	9·04	2·2017
7·79	2·05284	8·21	2·10535	8·63	2·15524	9·05	2·2028
7·80	2·05412	8·22	2·10657	8·64	2·15640	9·06	2·2039
7·81	2·05540	8·23	2·10778	8·65	2·15755	9·07	2·2050
7·82	2·05668	8·24	2·10899	8·66	2·15871	9·08	2·2061
7·83	2·05796	8·25	2·11021	8·67	2·15986	9·09	2·2072
7·84	2·05923	8·26	2·11142	8·68	2·16102	9·10	2·2083
7·85	2·06051	8·27	2·11263	8·69	2·16217	9·11	2·2094
7·86	2·06178	8·28	2·11384	8·70	2·16332	9·12	2·2105
7·87	2·06305	8·29	2·11504	8·71	2·16447	9·13	2·2116
7·88	2·06432	8·30	2·11625	8·72	2·16561	9·14	2·2127
7·89	2·06559	8·31	2·11745	8·73	2·16676	9·15	2·2138
7·90	2·06686	8·32	2·11866	8·74	2·16791	9·16	2·2148
7·91	2·06812	8·33	2·11986	8·75	2·16905	9·17	2·2159
7·92	2·06939	8·34	2·12106	8·76	2·17019	9·18	2·2170
7·93	2·07065	8·35	2·12226	8·77	2·17133	9·19	2·2181
7·94	2·07191	8·36	2·12345	8·78	2·17247	9·20	2·2192
7·95	2·07317	8·37	2·12465	8·79	2·17361	9·21	2·2203
7·96	2·07442	8·38	2·12584	8·80	2·17475	9·22	2·2214
7·97	2·07568	8·39	2·12704	8·81	2·17588	9·23	2·2225
7·98	2·07693	8·40	2·12823	8·82	2·17702	9·24	2·2235
7·99	2·07819	8·41	2·12942	8·83	2·17815	9·25	2·2246
8·00	2·07944	8·42	2·13060	8·84	2·17928	9·26	2·2257
8·01	2·08069	8·43	2·13179	8·85	2·18041	9·27	2·2268
8·02	2·08193	8·44	2·13298	8·86	2·18154	9·28	2·2279
8·03	2·08318	8·45	2·13416	8·87	2·18267	9·29	2·2289
8·04	2·08442	8·46	2·13534	8·88	2·18380	9·30	2·2300
8·05	2·08567	8·47	2·13653	8·89	2·18492	9·31	2·2311
8·06	2·08691	8·48	2·13771	8·90	2·18605	9·32	2·2322
8·07	2·08815	8·49	2·13888	8·91	2·18717	9·33	2·2332
8·08	2·08939	8·50	2·14006	8·92	2·18829	9·34	2·2343
8·09	2·09062	8·51	2·14124	8·93	2·18941	9·35	2·2354
8·10	2·09186	8·52	2·14241	8·94	2·19053	9·36	2·2364
8·11	2·09309	8·53	2·14358	8·95	2·19165	9·37	2·2375
8·12	2·09433	8·54	2·14476	8·96	2·19277	9·38	2·2386
8·13	2·09556	8·55	2·14593	8·97	2·19388	9·39	2·2396
8·14	2·09679	8·56	2·14710	8·98	2·19499	9·40	2·2407

TABLE A. (*Continued*)

No.	Logarithm	No.	Logarithm	No.	Logarithm	No.	Logarithm
9·41	2·2418	9·56	2·2576	9·71	2·2732	9·86	2.2885
9·42	2·2428	9·57	2·2586	9·72	2·2742	9·87	2·2895
9·43	2·2439	9·58	2·2597	9·73	2·2752	9·88	2·2905
9·44	2·2450	9·59	2·2607	9·74	2·2762	9·89	2·2915
9·45	2·2460	9·60	2·2618	9·75	2·2773	9·90	2·2925
9·46	2·2471	9·61	2·2628	9·76	2·2783	9·91	2·2935
9·47	2·2481	9·62	2·2638	9·77	2·2793	9·92	2·2946
9·48	2·2492	9·63	2·2649	9·78	2·2803	9·93	2·2956
9·49	2·2502	9·64	2·2659	9·79	2·2814	9·94	2·2966
9·50	2·2513	9·65	2·2670	9·80	2·2824	9·95	2·2976
9·51	2·2523	9·66	2·2680	9·81	2·2834	9·96	2·2986
9·52	2·2534	9·67	2·2690	9·82	2·2844	9·97	2·2996
9·53	2·2544	9·68	2·2701	9·83	2·2854	9·98	2·3006
9·54	2·2555	9·69	2·2711	9·84	2·2865	9·99	2·3016
9·55	2·2565	9·70	2·2721	9·85	2·2875	10·00	2·3026

TABLE B.—MEAN OR AVERAGE PRESSURE OF EXPANSIVE STEAM

	Points of stroke when steam is cut off.							
Cut off at	$\frac{1}{10}$	$\frac{1}{8}$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{3}{10}$	$\frac{1}{3}$	$\frac{4}{10}$
Number of expansions.	10	8	6	5	4	3·33	3	2·5
Hyp. log. (n + 1).	3·303	3·079	2·792	2·609	2·386	2·203	2·099	1·916
Initial pressure in lbs. per sq. inch.	Average pressure of steam in lbs. per square inch throughout the stroke.							
5	1·651	1·924	2·325	2·609	2·982	3·304	3·484	3·832
10	3·303	3·848	4·648	5·218	5·965	6·609	6·989	7·664
15	4·954	5·772	6·980	7·827	8·947	9·913	10·49	11·49
20	6·606	7·697	9·297	10·43	11·93	13·21	13·98	15·32
25	8·255	9·620	11·63	13·04	14·91	16·52	17·48	19·16
30	9·909	11·54	13·96	15·65	17·90	19·82	20·99	22·99
35	11·56	13·47	16·28	18·26	20·87	23·13	24·48	26·82
40	13·21	15·39	18·59	20·87	23·86	26·43	27·98	30·65
45	14·86	17·32	20·94	23·48	26·84	29·74	31·48	34·38
50	16·51	19·24	23·26	26·09	29·82	33·04	34·96	38·32
55	18·16	21·16	25·59	28·69	32·80	36·38	38·48	42·15
60	19·81	23·09	27·92	31·31	35·79	39·65	41·98	45·98
65	21·46	25·01	30·24	33·91	38·77	44·00	45·47	49·81
70	23·10	26·94	32·55	36·53	41·75	46·26	48·96	53·65

75	24·77	28·86	34·90	39·13	44·73	49·61	52·47	57·48
80	26·42	30·79	37·21	41·74	47·72	52·87	55·96	61·31
85	28·07	32·71	39·55	44·35	50·57	56·23	59·47	65·14
90	29·73	34·64	41·88	46·96	53·68	59·48	62·97	68·97
95	31·37	36·56	44·20	49·57	56·66	62·84	66·46	72·80
100	33·03	38·48	46·51	52·18	59·65	66·08	69·92	76·64
105	34·68	40·41	48·56	54·78	62·63	69·46	73·46	80·47
110	36·33	42·34	51·17	57·40	65·61	72·70	76·94	84·30
115	37·98	44·26	53·51	60·00	68·59	76·07	80·46	88·13
120	39·64	46·18	55·84	62·62	71·58	79·31	83·96	91·96
125	41·28	48·10	58·16	65·22	74·56	82·69	87·45	95·80
130	42·92	50·03	60·47	67·83	77·54	85·91	90·92	99·63
135	44·59	51·95	62·82	70·44	80·52	89·31	94·45	103·4
140	46·23	53·88	65·13	73·05	83·51	92·52	97·94	107·3
145	47·89	55·80	67·47	75·66	86·49	95·92	101·4	111·1
150	49·54	57·73	69·80	78·27	89·47	99·13	104·9	114·9
155	51·19	59·65	72·13	80·87	92·45	102·5	108·4	118·7
160	52·84	61·58	74·45	83·48	95·44	105·8	111·9	122·6
165	54·49	63·50	76·78	86·09	98·42	109·1	115·4	126·4
170	56·15	65·42	79·10	88·70	101·4	112·4	118·9	130·2
175	57·80	67·35	81·43	91·31	103·3	115·7	122·6	134·1
180	59·45	69·27	83·76	93·77	107·3	119·0	125·9	137·9
185	61·10	71·20	86·08	96·53	110·3	122·3	129·4	141·7
190	62·75	73·12	88·41	99·14	113·3	125·6	132·9	145·6
195	64·40	75·05	90·74	101·7	116·3	129·0	136·4	149·4
200	66·06	76·97	93·05	104·4	119·3	132·3	139·9	153·3

TABLE B. *(Continued)*

	Points of stroke when steam is cut off.							
Cut off at	$\frac{1}{2}$	$\frac{6}{10}$	$\frac{2}{3}$	$\frac{7}{10}$	$\frac{3}{4}$	$\frac{8}{10}$	$\frac{7}{8}$	$\frac{9}{10}$
Number of expansions.	2	1·66	1·5	1·43	1·33	1·25	1·14	1·11
Hyp. log. (n + 1).	1·693	1·507	1·405	1·357	1·284	1·223	1·131	1·104
Initial pressure in lbs. per sq. inch.	**Average pressure of steam in lbs. per square inch throughout the stroke.**							
5	4·232	4·518	4·683	4·745	4·827	4·892	4·960	4·973
10	8·465	9·037	9·366	9·489	9·654	9·784	9·921	9·946
15	12·69	13·55	14·05	14·23	14·48	14·67	14·88	14·92
20	16·93	18·07	18·73	18·98	19·30	19·56	19·84	19·89
25	21·16	22·59	23·41	23·72	24·13	24·46	24·80	24·86
30	25·39	27·11	28·10	28·46	28·96	29·35	29·76	29·84
35	29·64	31·62	32·78	33·21	33·79	34·24	34·72	34·81
40	33·86	36·15	37·43	37·95	38·61	39·13	39·68	39·78
45	38·09	40·66	42·14	42·70	43·44	44·02	44·64	44·75
50	42·32	45·18	46·82	47·44	48·27	48·92	49·60	49·73
55	46·55	49·93	51·51	52·19	53·09	53·81	54·56	54·70
60	50·79	54·22	56·20	56·93	57·92	58·70	59·52	59·67
65	55·02	59·00	60·88	61·68	62·75	63·59	64·48	64·64
70	**59·25**	**63·26**	**65·56**	**66·42**	**67·57**	68·48	**69·44**	**69·62**

75	63·48	68·08	70·25	71·17	72·40	73·38	74·40	74·59
80	67·72	72·30	74·86	75·91	77·23	78·27	79·37	79·56
85	71·95	77·16	79·61	80·66	82·06	83·16	84·32	84·54
90	76·18	81·34	84·28	85·40	86·88	88·05	89·28	89·50
95	80·41	86·24	88·98	90·15	91·71	92·94	94·25	94·48
100	84·65	90·36	93·64	94·89	96·54	97·84	99·21	99·46
105	88·88	95·32	98·35	99·63	101·3	102·7	104·1	104·4
110	93·11	99·40	103·0	104·4	106·2	107·6	109·1	109·4
115	97·34	104·4	107·7	109·1	111·0	112·5	114·0	114·3
120	101·6	108·4	112·4	113·8	115·8	117·4	119·0	119·3
125	105·8	113·4	117·0	118·6	120·6	122·3	124·0	124·3
130	110·0	117·5	121·7	123·3	125·5	127·2	128·9	129·3
135	114·2	122·5	126·0	128·1	130·3	132·0	133·9	134·2
140	118·5	126·5	131·1	132·8	135·1	136·9	138·8	139·2
145	122·7	131·6	135·8	137·5	139·9	141·8	143·8	144·2
150	126·9	135·5	140·5	142·3	144·8	146·7	148·8	149·2
155	131·2	140·7	145·5	147·0	149·6	151·6	153·7	154·1
160	135·4	145·2	149·9	151·8	154·4	156·5	158·7	159·1
165	139·6	149·7	154·5	156·5	159·2	161·4	163·6	164·1
170	143·9	154·3	159·2	161·3	164·1	166·3	168·6	169·0
175	148·1	158·9	163·9	166·0	168·9	171·2	173·6	174·0
180	152·8	163·4	168·6	170·8	173·7	176·1	178·5	179·0
185	156·6	167·9	173·2	175·5	178·6	181·0	183·5	184·0
190	160·8	172·4	177·3	180·3	183·4	185·9	188·5	188·9
195	165·0	177·0	182·6	185·0	188·2	192·3	193·4	193·9
200	169·3	180·7	187·3	189·8	193·0	195·6	198·4	198·9

TABLE C.—PROPERTIES OF SATURATED STEAM

Total pressure in lbs. per square inch.	Temperature in Fahr. degrees.	Total heat in B.Th.U. from water at 32°Fahr.	Latent heat in B.Th.U.	Density or weight of one cubic foot	Volume of one lb. of steam, cubic foot.
1	101·8	1104·4	1034·6	0·0030	333·0
2	126·1	1115·0	1021·0	0·0058	173·5
3	141·5	1121·6	1012·3	0·0084	118·5
4	153·0	1126·5	1005·1	0·0111	90·5
5	162·3	1130·5	1000·3	0·0136	73·33
6	170·0	1133·7	995·8	0·0162	61·89
7	176·8	1136·5	991·8	0·0187	53·56
8	182·9	1139·0	988·2	0·0211	47·27
9	188·3	1141·1	985·0	0·0236	42·36
10	193·2	1143·1	982·0	0·0261	38·38
11	197·7	1144·9	979·2	0·0285	35·10
12	202·0	1146·5	976·6	0·0309	32·36
13	205·9	1148·0	974·2	0·0333	30·03
14	209·5	1149·4	971·9	0·0357	28·02
14·7	212·0	1150·4	970·4	0·0373	26·79
15	213·0	1150·7	969·7	0·0381	26·27
16	216·3	1152·0	967·6	0·0404	24·79
17	219·4	1153·1	965·6	0·0428	23·38
18	222·4	1154·2	963·7	0·0451	22·16
19	225·2	1155·2	961·8	0·0475	21·07
20	228·0	1156·2	960·0	0·0498	20·08
21	230·6	1157·1	958·3	0·0521	19·18
22	233·1	1158·0	956·7	0·0544	18·37
23	235·5	1158·8	955·1	0·0568	17·62
24	237·8	1159·6	953·5	0·0591	16·93
25	240·1	1160·4	952·0	0·0614	16·30
26	242·2	1161·2	950·6	0·0636	15·72
27	244·4	1161·9	949·2	0·0659	15·18
28	246·4	1162·6	947·8	0·0682	14·67
29	248·4	1163·2	946·4	0·0705	14·19
30	250·3	1163·9	945·1	0·0728	13·74
31	252·2	1164·5	943·8	0·0751	13·32
32	254·1	1165·1	942·5	0·0773	12·93
33	255·8	1165·7	941·3	0·0795	12·57
34	257·6	1166·3	940·1	0·0818	12·22
35	259·3	1166·8	938·9	0·0841	11·89
36	261·0	1167·3	937·7	0·0863	11·58
37	262·6	1167·8	936·6	0·0886	11·29

TABLE C. (*Continued*)

Total pressure in lbs. per square inch.	Temperature in Fahr. degrees.	Total heat in B.Th.U. from water at 32°Fahr.	Latent heat in B.Th.U.	Density or weight of one cubic foot.	Volume of one lb. of steam, cubic foot.
38	264·2	1168·4	935·5	0·0908	11·01
39	265·8	1168·9	934·4	0.0931	10·74
40	267·3	1169·4	933·3	0·0953	10·49
41	268·7	1169·8	932·2	0·0976	10·25
42	270·2	1170·3	931·2	0·0998	10·02
43	271·7	1170·7	930·2	0·1020	9·80
44	273·1	1171·2	929·2	0·1043	9·59
45	274·5	1171·6	928·2	0·1065	9·39
46	275·8	1172·0	927·2	0·1087	9·20
47	277·2	1172·4	926·3	0·1109	9·02
48	278·5	1172·8	925·3	0·1131	8·84
49	279·8	1173·2	924·4	0·1153	8·67
50	281·0	1173·6	923·5	0·1175	8·51
51	282·3	1174·0	922·6	0·1197	8·35
52	283·5	1174·3	921·7	0·1219	8·20
53	284·7	1174·7	920·8	0·1241	8·05
54	285·9	1175·0	919·9	0·1263	7·91
55	287·1	1175·4	919·0	0·1285	7·78
56	288·2	1175·7	918·2	0·1307	7·65
57	289·4	1176·0	917·4	0·1329	7·52
58	290·5	1176·4	916·5	0·1350	7·40
59	291·6	1176·7	915·7	0·1372	7·28
60	292·7	1177·0	914·9	0·1394	7·17
61	293·8	1177·3	914·1	0·1416	7·06
62	294·9	1177·6	913·3	0·1438	6·95
63	295·9	1177·9	912·5	0·1460	6·85
64	297·0	1178·2	911·8	0·1482	6·75
65	298·0	1178·5	911·0	0·1503	6·65
66	299·0	1178·8	910·2	0·1525	6·56
67	300·0	1179·0	909·5	0·1547	6·47
68	301·0	1179·3	908·7	0·1569	6·38
69	302·0	1179·6	908·0	0·1590	6·29
70	302·9	1179·8	907·2	0·1612	6·20
71	303·9	1180·1	906·5	0·1634	6·12
72	304·8	1180·4	905·8	0·1656	6·04
73	305·8	1180·6	905·1	0·1678	5·96
74	306·7	1180·9	904·4	0·1699	5·89
75	307·6	1181·1	903·7	0·1721	5·81

TABLE C. (*Continued*)

Total pressure in lbs. per square inch.	Temperature in Fahr. degrees.	Total heat in B.Th.U. from water at 32°Fahr.	Latent heat in B.Th.U.	Density or weight of one cubic foot.	Volume of one lb. of steam, cubic foot.
76	308·5	1181·4	903·0	0·1743	5·74
77	309·4	1181·6	902·3	0·1764	5·67
78	310·3	1181·8	901·7	0·1786	5·60
79	311·2	1182·1	901·0	0·1808	5·54
80	312·0	1182·3	900·3	0·1829	5·47
81	312·9	1182·5	899·7	0·1851	5·41
82	313·8	1182·8	899·0	0·1873	5·34
83	314·6	1183·0	898·4	0·1894	5·28
84	315·4	1183·2	897·7	0·1915	5·22
85	316·3	1183·4	897·1	0·1937	5·16
86	317·1	1183·6	896·4	0·1959	5·10
87	317·9	1183·8	895·8	0·1980	5·05
88	318·7	1184·0	895·2	0·2001	5·00
89	319·5	1184·2	894·6	0·2023	4·94
90	320·3	1184·4	893·9	0·2044	4·89
91	321·1	1184·6	893·3	0·2065	4·84
92	321·8	1184·8	892·7	0·2087	4·79
93	322·6	1185·0	892·1	0·2109	4·74
94	323·4	1185·2	891·5	0·2130	4·69
95	324·1	1185·4	890·9	0·2151	4·65
96	324·9	1185·6	890·3	0·2172	4·60
97	325·6	1185·8	889·7	0·2193	4·56
98	326·4	1186·0	889·2	0·2215	4·51
99	327·1	1186·2	888·6	0·2237	4·47
100	327·8	1186·3	888·0	0·2258	4·43
101	328·6	1186·5	887·4	0·2279	4·39
102	329·3	1186·7	886·9	0·2300	4·35
103	330·0	1186·9	886·3	0·2322	4·31
104	330·7	1187·0	885·8	0·2343	4·27
105	331·4	1187·2	885·2	0·2365	4·23
106	332·0	1187·4	884·7	0·2386	4·19
107	332·7	1187·5	884·1	0·2408	4·15
108	333·4	1187·7	883·6	0·2429	4·12
109	334·1	1187·9	883·0	0·2450	4·08
110	334·8	1188·0	882·5	0·2472	4·05
111	335·4	1188·2	881·9	0·2493	4·01
112	336·1	1188·4	881·4	0·2514	3·98
113	336·8	1188·5	880·9	0·2535	3·94

Table C. (*Continued*)

Total pressure in lbs. per square inch.	Temperature in Fahr. degrees.	Total heat in B.Th.U. from water at 32° Fahr.	Latent heat in B.Th.U.	Density or weight of one cubic foot.	Volume of one lb. of steam, cubic foot.
114	337·4	1188·7	880·4	0·2556	3·912
115	338·1	1188·8	879·8	0·2577	3·880
116	338·7	1189·0	879·3	0·2599	3·848
117	339·4	1189·1	878·8	0·2620	3·817
118	340·0	1189·3	878·3	0·2641	3·786
119	340·6	1189·4	877·8	0·2662	3·756
120	341·3	1189·6	877·2	0·2683	3·726
121	341·9	1189·7	876·7	0·2705	3·697
122	342·5	1189·8	876·2	0·2726	3·668
123	343·2	1190·0	875·7	0·2748	3·639
124	343·8	1190·1	875·2	0·2769	3·611
125	344·4	1190·3	874·7	0·2791	3·583
126	345·0	1190·4	874·2	0·2812	3·556
127	345·6	1190·5	873·8	0·2833	3·530
128	346·2	1190·7	873·3	0·2854	3·504
129	346·8	1190·8	872·8	0·2875	3·478
130	347·4	1191·0	872·3	0·2897	3·452
131	348·0	1191·1	871·8	0·2918	3·427
132	348·5	1191·2	871·3	0·2939	3·402
133	349·1	1191·3	870·9	0·2960	3·378
134	349·7	1191·5	870·4	0·2981	3·354
135	350·3	1191·6	869·9	0·3002	3·331
136	350·8	1191·7	869·4	0·3023	3·308
137	351·4	1191·8	869·0	0·3044	3·285
138	352·0	1192·0	868·5	0·3065	3·263
139	352·5	1192·1	868·1	0·3086	3·241
140	353·1	1192·2	867·6	0·3107	3·219
141	353·6	1192·3	867·2	0·3129	3·197
142	354·2	1192·5	866·7	0·3150	3·175
143	354·7	1192·6	866·3	0·3171	3·154
144	355·3	1192·7	865·8	0·3192	3·133
145	355·8	1192·8	865·4	0·3213	3·112
146	356·3	1192·9	864·9	0·3234	3·092
147	356·9	1193·0	864·5	0·3255	3·072
148	357·4	1193·2	864·0	0·3276	3·052
149	357·9	1193·3	863·6	0·3297	3·033
150	358·5	1193·4	863·2	0·3320	3·012
155	361·0	1194·0	861·0	0·3425	2·920

TABLE C. (*Continued*)

Total pressure in lbs. per square inch.	Temperature in Fahr. degrees.	Total heat in B.Th.U. from water at 32°Fahr.	Latent heat in B.Th.U.	Density or weight of one cubic foot.	Volume of one lb. of steam, cubic foot.
160	363·6	1194·5	858·8	0·3529	2·834
165	366·0	1195·0	856·8	0·3633	2·753
170	368·5	1195·4	854·7	0·3738	2·675
175	370·8	1195·9	852·7	0·3843	2·602
180	373·1	1196·4	850·8	0·3948	2·533
185	375·4	1196·8	848·8	0·4052	2·468
190	377·6	1197·3	846·9	0·4157	2·406
195	379·8	1197·7	845·4	0·4262	2·346
200	381·9	1198·1	843·2	0·437	2·290
205	384·0	1198·5	841·4	0·447	2·237
210	386·0	1198·8	839·6	0·457	2·187
215	388·0	1199·2	837·9	0·468	2·138
220	389·9	1199·6	836·2	0·478	2·091
225	391·9	1199·9	834·4	0·489	2·046
230	393·8	1200·2	832·8	0·499	2·004
235	395·6	1200·6	831·1	0·509	1·964
240	397·4	1200·9	829·5	0·520	1·924
245	399·3	1201·2	827·9	0·530	1·887
250	401·1	1201·5	826·3	0·541	1·850
260	404·5	1202·1	823·1	0·561	1·782
270	407·9	1202·6	820·1	0·582	1·718
280	411·2	1203·1	817·1	0·603	1·658
290	414·4	1203·6	814·2	0·624	1·602
300	417·5	1204·1	811·3	0·645	1·551
310	420·5	1204·5	808·5	0·666	1·502
320	423·4	1204·9	805·8	0·687	1·456
330	426·3	1205·3	803·1	0·708	1·413
340	429·1	1205·7	800·4	0·729	1·372
350	431·9	1206·1	797·8	0·750	1·334
400	444·7	1208	786	0·86	1·17
450	456·5	1209	774	0·96	1·04
500	467·2	1210	762	1·08	0·93
550	477·2	1210	752	1·18	0·85
600	486·4	1211	742	·28	0·78

FUELS, STEAM, STEAM GENERATORS

FUELS—THEORY OF COMBUSTION

COMBUSTION is essentially rapid chemical combination with the oxygen of the air accompanied by the evolution of heat and light. The general theory of combustion is best explained in the first place by taking pure carbon (C), the essential basis of most fuels.

Air contains roughly 21 per cent. oxygen and 79 per cent. nitrogen by volume; the other normal constituents of air, such as carbon dioxide, ozone, and ammonia, are present in such minute quantities that they can be ignored in considering the theory of combustion.

In the combustion of carbon, the latter is first heated up to the temperature of ignition (about 1240° F.), and then "burns," that is to say it combines with the oxygen of the air, giving off light and heat, and forms a colourless gas, carbon dioxide CO_2, expressed in a chemical equation as $C + O_2 = CO_2$.

In this process, 1 lb. of pure carbon forms 3·66 lbs. of carbon dioxide gas, and evolves approximately 14,600 British Thermal Units of Heat. This combustion really takes place in two stages, namely, first, the formation of carbon monoxide gas (CO), $2C + O_2 = 2CO$, in which 1 lb. of carbon forms 2·33 lbs. of carbon monoxide and evolves 4,400 B.Th.U.; and, secondly, this carbon monoxide, being an inflammable gas, then burns to CO_2, $CO + O = CO_2$, giving 3·66 lbs. of carbon dioxide, with the evolution of approximately 10,200 B.Th.U. If the supply of air is insufficient, then the products of combustion will be a mixture of CO_2 and CO depending on the proportions of carbon and air present. It will be realized, therefore, that if any portion of carbon burns to CO only which escapes unburnt, then the heat loss will be very serious.

Fuels, of course, contain other combustible elements besides carbon, chiefly hydrogen and sulphur.

In the combustion of hydrogen, 1 lb. of hydrogen heated up to the temperature of ignition (approximately 1070° F.) combines with the oxygen of the air to give 9·00 lbs. of water which escapes from the fire as steam at the temperature of the exit gases, say, 350° F., and evolves approximately 52,800 B.th.u. of heat net ($2H_2 + O_2 = 2H_2O$). The actual amount of heat evolved is complicated by the fact that the water produced abstracts some heat to form steam at atmospheric pressure. However, 1 lb. of hydrogen evolves about 61,500 B.Th.U. If now the fuel (and the water in the fuel) is taken as 32° F., then to heat the water formed to 212° F. and convert it to steam at this temperature, then heat is abstracted, and the net heat evolved is about 51,700 B.Th.U.

In the combustion of sulphur, 1 lb. of sulphur heated up to the temperature of ignition (685° F.) combines with the oxygen of the air to give 2·00 lbs. of sulphur dioxide gas ($S + O_2 = SO_2$), and evolves approximately 4,000 B.Th.U.

It will be readily understood that the chemical reactions taking place in the combustion of any given coal are of a very complicated character. Taking ordinary steam coal, however, with a simple average composition of, say, 80·0 per cent. carbon, 5·0 per cent. hydrogen, 7·5 per cent. oxygen, 2·5 per cent. sulphur, 1·0 per cent. nitrogen, and 4·0 per cent. mineral matter. This complicated fuel is heated up to the ignition temperature of coal, which, according to Vivian Lewes, varies from 668° F. in the case of cannel coal to 925° F. in the case of anthracite, and then combines with the oxygen of the air, that is to say, it "burns." The carbon burns, as already described, to CO_2 with some CO if there is not sufficient air, the sulphur burns to SO_2, the nitrogen escapes uncombined, water is formed by the combination of the hydrogen and escapes as steam, and the mineral matter remains behind as ash. The net result is the evolution of, say, 13,500 B.Th.U. of heat per lb., and the same general theory applies to the combustion of all fuels.

In practice, considerable excess air has to be used to burn any given fuel, because of the difficulty of bringing the air into intimate contact with every part of the fuel. Thus, for example, the theoretical amount of air for the combustion of 1 lb. of average steam coal is about 11·0 lbs.; for Welsh steam coal, say, 11·5 lbs.; and for anthracite, say, 11·75 lbs., as can be calculated from the chemical formulae already given. Thus 1 lb. of pure carbon would require about 11·5 lbs. of air equal to, say, 142 cub. ft.

The amount of air required can be expressed by the following approximate formulae—

$$\text{lbs. of air required per lb. of coal} = 11{\cdot}5\,C + 34{\cdot}8D\left(H - \frac{0}{8}\right)$$

where C = % of carbon
H = % of hydrogen
O = % of oxygen

In practice, the actual amount of air required with chimney draught is about 19–22 lbs., that is, a very large excess (65–90 per cent.) of, say, 11·5 lbs. theoretical for average quality coal, and representing a considerable loss of efficiency due to the heat carried away by the excess of air.

The theoretical temperature of combustion of coal of a given quality can be calculated, and will be found for average quality of coal to be about 4,500° F. In practice, however, this is considerably reduced by the excess air necessary. Thus, 100 per cent. excess air would give a calculated temperature of only about 2,500° F.

CLASSIFICATION OF FUELS.

The fuels in use for steam generation may be classified as follows—

A. *Solid Fuels.*

Coal (together with powdered coal, "coalite" and other similar residues of low temperature carbonization, coke and briquettes).

Lignite (brown coal)
Peat
Wood
Bagasse (sugar-cane residue)
Spent tan (tanyard refuse)
General town and factory refuse

B. *Liquid Fuels.*

Petroleum (oil fuel)
Shale oil (oil fuel)
Tar

C. *Gaseous Fuels.*

Blast furnace gas
Producer gas
Water gas
Coal gas
Natural gas
Waste heat

SOLID FUELS

1. COAL STATISTICS.

Coal is, of course, the essential fuel for steam generation. The amount of coal used in the world is enormous, and the figures are advancing by leaps and bounds. Thus—

World's Coal Production
(Metric tons).

1800	11,600,000	1870	213,400,000
1820	17,200,000	1880	340,100,000
1830	25,100,000	1890	466,100,000
1840	44,800,000	1900	800,200,000
1850	81,400,000	1910	1,141,600,000
1860	142,300,000	1916	1,345,100,000

The amount of coal produced per annum by different countries is as follows, taking the year 1913, and expressed in metric tons—

United States	569,000,000	Russia .	35,000,000
Great Britain	321,000,000	Belgium .	25,000,000
Germany .	305,000,000	Japan .	23,000,000
Austria-Hungary	59,000,000		
France . . .	45,000,000	Total	1,382,000,000

The following is the estimated present coal reserves of the principal coal-producing countries of the world, expressed as metric tons. On this assumption, the coalfields of Great Britain will be exhausted in about 500 years—

United States	3,527,000,000,000	Japan	50,000,000,000
China ..	1,500,000,000,000	Austria-Hungary	30,000,000,000
Great Britain	180,000,000,000	France	25,000,000,000
Germany ..	164,000,000,000	Belgium ..	20,000,000,000
Canada ..	100,000,000,0000	Chile	2,000,000,000

According to the recent interim report on electric power supply in Great Britain of the Reconstruction Committee (Coal Conservation Sub-Committee) Ministry of Reconstruction, Cd. 9084, the total annual output of coal of all the mines of the United Kingdom before the war (taking 1913 as typical) was approximately 287,000,000 tons, of which 189,000,000 tons were consumed at home as follows—

		TONS.
1.	Railways	15,000,000
2.	Coasting steamers	2,500,000
3.	Factories	60,000,000
4.	Mines	20,500,000
5.	Iron and steel industries	31,000,000
6.	Other metals and minerals..	1,250,000
7.	Brickworks, potteries, glass works, chemical works	5,750,000
8.	Gasworks	18,000,000
9.	Domestic	35,000,000
		189,000,000

The Author estimates ("Coal Saving by the Scientific Control of Steam Boiler Plants," *Engineering*, 12–19th July, 1918) that of this total amount of 189,000,000 tons per annum, 75–100,000,000 tons, that is 40–53 per cent., is used for steam generation alone in stationary land boilers.

According to the United States Geological Survey of 1915, out of a total consumption (bituminous and anthracite coal) of approximately 532,000,000 metric tons (2,000 lbs.), approximately 185,000,000 tons, that is 35 per cent., was used for steam generation only in stationary land boilers. In the United States a very big percentage is used for railways, approximately 128,000,000 tons, that is 24 per cent.; whilst in Great Britain, as seen above, the figure is only 15,000,000 ordinary tons, or about 8 per cent.

COAL

FORMATION AND COMPOSITION.

Coal has been formed from vegetable matter which has been subjected for millions of years to the action of pressure, heat, and bacteria in the interior of the earth. The approximate figures for the composition of the original vegetable matter from which coal has been formed are, say, 50·0 per cent. carbon, 6·0 per cent. hydrogen, 43·5 per cent. oxygen, traces of nitrogen, and 0·5 per cent. mineral matter (ash) in the dried matter. Under natural conditions, however, the percentage of water may have been anything from 30–60 per cent.

During the formation of coal, this original vegetable matter gradually lost its water and then the oxygen and hydrogen, so that the percentage of carbon increased according to the age or "rank" of the coal.

The first stage may be said to be that of peat or analogous compounds, where the composition is, say, 60·0 per cent. carbon, 6·0 per cent. hydrogen, and 33·5 per cent. oxygen.

The next stage is that of lignite or brown coal. There are many varieties of lignite, depending on the "rank"; but, roughly, an average composition is, say, 66·0 per cent. carbon, 5·5 per cent. hydrogen, and 28·0 per cent. oxygen.

The following stage is that of coal proper, of which there are many varieties, depending again on the "rank" or age. An average bituminous coal can be taken as 82–85 per cent. carbon, 4·5–5·5 per cent. hydrogen, and 4·0 per cent. oxygen. A semi-bituminous coal will be, say, 89–90 per cent. carbon, 4·0–5·0 per cent. hydrogen, and 2–4 per cent. oxygen. The final stage is that of anthracite coal. Thus semi-anthracite contains 91–93 per cent. carbon, 3–4·5 per cent. hydrogen, and 3·5 per cent. oxygen. Finally, we have anthracite proper, containing 92·5–95 per cent. carbon, less than 4·0 per cent. hydrogen, and less than 3·0 per cent. oxygen, a hard, shiny black smokeless coal—so hard that it will not mark paper. Presumably the percentage of carbon would continue to increase if the anthracite was to remain longer in the earth.

As will be seen, therefore, coal is a substance of varying "rank" and composition, but it consists essentially of carbon, with different amounts of hydrogen, oxygen, sulphur, nitrogen, and mineral matter (ash). It has a specific gravity of from 1·2 to 1·5, and varies considerably in texture and hardness. Ordinary steam coal is about 45–50 cub. ft. to the ton, and Welsh steam coal 40–42 cub. ft.

The moisture content may be anything from 1–10 per cent. and the heating value, say, 11,000 – 15,000 B.Th.U. per lb.

Coal is roughly classified as follows (Prof. J. S. Brame), but there is no sharp dividing line between any of the proposed divisions.

TABLE I. COMPOSITION OF COALS

	C.	H.	O.	Volatile Matter.
1. Lignitious	75–80	4·8–5·5	12–30	35–47
2. Ligno-bituminous ..	78–84	4·5–6·0	8–13·5	35–45
3. Bituminous long flame (non-caking)	82–86	5·0–6·0	6–12	30–40
4. Bituminous long flame (partly caking) ..	82–86	4·5–5·5	5–9	30 40
5. Bituminous short flame coking	85–89	4·5–5·5	4–7·5	20–30
6. Semi-bituminous ..	89–92	4·0–5·0	2–4·5	13–20
7. Semi-anthracite ..	91–93	3·0–4·5	3–5	8–13
8. Anthracite..	over 92·5	less than 4·0	less than 3	less than 8

Coal is also distinguished as "caking" and "non-caking." "Caking" coals soften when heated, and yield a coherent mass of spongy coke; "non-caking" coals burn without softening, and do not form a coherent coke.

Another rough division depends on the relative amount of "fixed" carbon" and "volatile matter." Thus anthracite coal contains 90–95 per cent. fixed carbon and 1–4 per cent.

volatile matter. Semi-anthracite ("steam" coal of South Wales), 70–80 per cent. fixed carbon and 10–15 per cent. volatile matter. Ordinary bituminous coal contains 50–70 per cent. fixed carbon and 25–35 per cent. volatile matter. Cannel coal is a variety of coal very high in volatile matter, containing over 35 per cent.

Ordinary steam coal as purchased for boiler-firing is, of course, of many different qualities depending on the quality of the original seam and the treatment it has received at the colliery. Coal from the pit face is riddled and graded, and may also be washed in washing machines and graded into "washed nuts," "washed beans," "washed peas," and so on. A good washed semi-bituminous slack may contain 12,500–13,500 B.Th.U. and 7–8 per cent. ash, whilst rough coal from the pit face will be, say, 12,000 B.Th.U. and 10–12 per cent. ash.

The Author found that the average analysis of the coal used by 250 different boiler plants during the last ten years, and representing an annual coal bill of 2,166,000 tons of coal, was 11,822 B.Th.U. per lb. and 11·5 per cent. ash.

Many inferior grades of coal are burnt under steam boilers, especially at collieries. Thus "washer settlings," the residue from the washing plants, may be of a quality of, say, 25–30 per cent. ash and 25 per cent. water, and containing 6,000–10,000 B.Th.U. per lb. Other inferior coal is mixed with large quantities of shale, and contains on this account up to 40 per cent. or even 50 per cent. ash. Another inferior coal extremely difficult to burn is fine anthracite and semi-anthracite, which may contain up to 12,000 B.Th.U., but only, say, 3–5 per cent. volatile matter.

The ash of coal generally consists of silicates of aluminium, iron and lime, say, 50·0 per cent. silica (SiO_2), and also to a small extent, sulphates. The iron and the sulphate is derived from iron pyrites found in the coal. Iron is obnoxious, as it tends to increase the fusibility of the ash. Ash generally fuses between 2,200–2,750° F. and coal with a low fusibility ash is very objectionable, as the ash "runs" and makes up the firebars.

PULVERIZED COAL

It is claimed that more economical results are obtained by burning coal in the form of a fine powder. There are various systems of carrying out this operation, but generally the rough coal is run through a preliminary crushing plant to break it down to ¾ in. mesh, so that it can be handled and dried. This coal is then conveyed to a pulveriser after passing through a magnetic separator to separate particles of iron. This crushed coal is then dried in special driers, coal-fired. The gases generally first heat the outer shell of the dryer and then towards the finish pass over the coal itself. The coal has to be dried to within 1·0 per cent. of moisture, and generally one passage through the drier will take out about 10 per cent. of water. If, therefore, the coal is very wet, it has to be dried in two stages.

The dried crushed coal is then pulverized in special pulverizing mills. The powdered coal has to be extraordinarily fine, say, 85 per cent. through a mesh with 40,000 holes to the square inch, and 95 per cent. through a mesh with 10,000 holes to the square inch.

The pulverized coal dust is then conveyed to the burners by various methods, such as compressed air, screw conveyors, cable pipe system, etc.

Very many types of coal dust burners are on the market, generally of the low-pressure compressed air type, and the coal dust falls into the main pipe taking the compressed air.

The following are the advantages and disadvantages of pulverized coal burning—

ADVANTAGES.

1. Flexibility of control of fuel and air, and ability to extinguish the fire instantly.

2. Complete combustion and no smoke. Ash shows no trace of combustible matter.

3. No clinker and "cleaning out."

4. Low-grade fuels can be burnt efficiently regardless of ash, sulphur, and other impurities.

5. Very little excess air and maximum fuel economy resulting in 16 per cent. CO_2.

6. Pulverized fuel has semi-liquid properties, flows easily, and can be transferred great distances through pipes.

7. Said to save 20–25 per cent. of coal bill.

8. Saving in labour and attendance ; also no stand by losses, damping and banking up.

9. Can take in an emergency an enormous overload, up to 100 per cent.

10. Pulverized fuel is claimed to be specially applicable to locomotive firing, marine work, open hearth, steel melting furnaces, forge furnaces, puddling furnaces, and soaking pits

DISADVANTAGES.

1. Danger of serious explosions due to the use of large quantities of extremely fine coal dust.

2. Rapid deterioration of furnace linings.

3. Absence of proper control of the temperature and amount of coal burnt to correspond to fluctuations in the steam demand.

4. Cost and trouble, together with risk of breakdowns, and difficulty with obnoxious dust, in grinding large quantities of coal to a fine powder and drying it to within 1 per cent. of moisture, with subsequent loss. According to makers' catalogues,* the total inclusive cost of handling the coal from lump to powder is about 40 per cent. of the price of the coal.

5. Serious danger of spontaneous combustion of finely-powdered coal in storage bins. Must not have more than forty-eight hours' supply. Also temperature of coal must not be over 200° F. for this reason.

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* *See* also *Pulverised Coal Systems in America* (L. C. Harvey), Department of Scientific and Industrial Research, H.M. Stationery Office (price 2s. 6d.), which gives a valuable bibliography in connection with powdered fuel.

6. Another great difficulty in connection with pulverized coal applied to steam generation is that the ash and molten slag is blown all over the boiler, and deposits especially on the tubes, filling up the combustion chamber, etc. It is claimed that this difficulty has now been got over, and a special boiler is not necessary. The ash difficulty has been practically solved by cutting down the volocity of hot gases in the combustion chamber to a minimum, the bulk of the ash being removed before the hot gases reach the boiler tubes.

COALITE*

There is no doubt that the burning of raw coal is a wasteful and unscientific process, as many valuable products, such as Diesel oil, fuel oil, motor spirit, and sulphate of ammonia are simply burnt as fuel. Now that the practical difficulties are being overcome, the low temperature carbonization of coal will be of the greatest importance.

The aim of low-temperature carbonization is to recover as much as possible of the by-products and to get a residual fuel that burns more easily than coke. The low-temperature carbonization process was invented by Thomas Parker, of Wolverhampton, and consists in submitting coal to a distillation process at a temperature of about 1000° F. It is carried out in flat rectangular retorts, placed vertically in a gas-fired furnace, and the distillation is completed roughly in 8 hrs.

YIELD FROM ONE TON OF AVERAGE COAL (say 25-30 per cent. volatile matter

	Low-temperature Carbonization.	High-temperature Carbonization.	
		Gasworks.	Coke Ovens.
Temperature of carbonization . .	1,000 deg. F. .	About 1,800 deg.F.	About 1,800 deg.F
Gas . . .	6,000-6,500 cubic feet of rich gas, 700-750 B.Th.U. per cubic foot.	12,000 cubic feet of medium quality (town's) gas, 550 B.Th.U. per cubic foot.	11,500 cubic feet of poor quality coke-oven gas, 450 B.Th.U. per cubic foot.
Liquid . . .	20 gal. coalite oil (fractionated to, say, 3 gals. motor spirit, 8-9 gal. Diesel or fuel oil, and 8-9 gal. lubricating oil).	10 gals. coal tar . Fractionated to about half-gallon motor spirit, carbolic acid, a large amount of naththalene, fuel oil, etc.	8 gals. coal tar
Sulphate of ammonia (from the ammonia)	15 lb. . . .	25 lb. . . .	28 lb. . . .
Residue in retort .	14 cwt. of smokeless fuel coalite containing about 9-10 per cent. volatile matter.	13½ cwt. of soft coke, containing about 1 per cent. volatile matter.	14-14½ cwt. of hard coke, containing less than ½ per cent. volatile matter.

* (Product of the low temperature carbonization of coal), as supplied by Messrs. The Low Temperature Carbonization, Ltd., 14–16 Cockspur Street, London, W.

The whole of the volatile products of the coal are not expelled as with coking. Each retort takes 10 cwts. of coal and yields approximately 7 cwts. of "coalite" (i.e. roughly 70 per cent. of the original coal, depending on the quality).

In comparison with ordinary gas manufacture, the figures are as shown on page 596.

Low-temperature carbonization reduces the yield of sulphate of ammonia, and the tar is much different in chemical constitution. Compared with ordinary coal tar, it contains no naphthalene or free carbon, but is much richer in light oils, benzene, toluol and solvent naphtha.

As a further example, the following are the actual figures supplied by Messrs. The Low Temperature Carbonization, Ltd., from tests carried out at their installation at Barugh, near Barnsley. The coal used was low-grade colliery slack, containing up to 22 per cent. ash and washed. After washing, the composition was Ash, 5·2 per cent.; volatile matter, 34·5 per cent.; fixed carbon, 54·3 per cent.; and water, 6·0 per cent.

The products of low-temperature carbonization of 1 ton of coal were—

	£	*s.*	*d.*
1. Gas, 7,000 ft. 640 B.Th.U. per cubic foot: Value 2s. per 1,000 ft.		14	–
2. Smokeless Fuel (Coalite), which is intermediate in quality between coke and coal, and claimed to be better than coal for household purposes. Value, 43s. per ton	1	10	–
3. Tar Oil: 17 galls. (for burning, lighting, and lubricating) for distilling and separation into numerous valuable organic compounds, or for burning under boilers. Value, 6d. per gallon		8	6
4. Motor Spirit: 3 galls., which, after treating, is valued at 1s. 8d. per gallon ..		5	–
5. Sulphate of Ammonia: 20 lbs. Value, say, £20 per ton less £8 for treatment ..		2	5
Total..	£2	19	11

The value of the coal used was 25s. per ton; and the total cost of treating 1 ton of coal, including wages, salaries, maintenance, rates and taxes, stationery and printing, miscellaneous interest on capital and depreciation on the plant at 15 per cent., is £2 7s. 1½d. per ton, leaving therefore a handsome profit.

The cost of the plant for low carbonization is, roughly, £700 per ton of coal treated for small plants, and £600 for large plants.

"Coalite," the residual fuel after the distillation, is intermediate in properties between coal and coke; and an average composition is 80 per cent. fixed carbon, 12·0 per cent. volatile matter, 1·0 per cent. sulphur, and 7·0 per cent. ash. It is stated by Professor Vivian Lewes to have an average heating value of 13,500 B.Th.U. per lb., and to burn, like coke, smokelessly and with the emission of much radiant heat.

COKE

Coke is the residual product obtained in the high temperature distillation of bituminous coal. Gasworks coke is obtained as a residue by the rapid heating of the coal in closed retorts, at a high temperature, in the process of coal-gas manufacture. As already seen, a ton of average gas-coal in this process yields about 66 per cent. of coke (1,480 lbs.), together with, say, 12,000 cub. ft. of coal-gas (14 c.-p.), 10 galls. of coal-tar, and 25 lbs. sulphate of ammonia.

Coke-oven coke is made in special coke-oven plants, in which the coal is slowly heated to a very high temperature. The residual coke is in a different physical condition to gasworks coke, being very much harder. It is employed chiefly in blast furnaces, and commands a higher price than gasworks coke.

In coke-ovens, the yield averages 50–65 per cent. coke, with 2·0–4·0 per cent. tar and 1–2 per cent. sulphate of ammonia.

Of course, all coals will produce coke; but many coals are not suitable for "coking," because the coke produced has no commercial value. Commercial coke is produced from the caking variety of bituminous coal. Many coals that will not coke well can be rendered quite suitable by crushing and washing, and mixing with other coals, so that the mixture is about 25 per cent. volatile matter. The finest coking coals in Great Britain are in the Durham coalfields.

Coke contains anything from 80–95 per cent. fixed carbon, 0·5–1·5 per cent. sulphur, and 5·5–16 per cent. ash. It must be clean, porous, and crystalline. Unfortunately, it readily absorbs moisture even up to 20 per cent., and this point is very important. A good sample of coke should not contain more than 5 per cent. moisture, and will then be about 12,500 B.Th.U. per lb., with, say, 10–12 per cent. ash. Sulphur (in the volatile state) and phosphorous are most objectionable. Coke of good quality, if it can be obtained at a reasonable price, is a most valuable fuel for steam generation. It burns easily with abundant radiant heat and no smoke, the fires are easily cleaned out, and very high efficiency obtained.

BRIQUETTES

Briquettes are generally made of any small or inferior coal mixed with some inflammable binding material to act as a cement.

If fine anthracite is used, it must be mixed with a certain proportion of fine bituminous coal, so as to be able to burn properly. Lignite can also be used, either alone or mixed with other coal, and sawdust is also often added.

The cementing material is generally pitch mixed with more or less rosin. The amount of pitch used depends on the nature of the coal, but is generally 4–10 per cent. by weight of the total briquette and, say, 1½ per cent. of rosin in addition. The pitch and rosin are ground and mixed with the powdered coal, and the mass is heated by steam until softened, and then pressed at about 1 ton per sq. in. into shapes in a briquetting machine. The shapes of the finished briquettes can be small, spherical or flat, like soap tablets, or in fairly large blocks for steam generation.

Many other substances have been used as cementing material, such as thick oil residues, Irish moss, clay, coal tar, and so on. Generally speaking, the more bituminous the fuel basis, the less cementing material is required.

A good briquette must be such that it does not become too soft in the fire whilst burning, and must not alter its shape in a hot climate. The heating value of such briquettes is generally, say, 7,400–8,500 B.Th.U. per lb.

LIGNITE

As already seen, lignite is intermediate in properties between woody matter and coal, that is to say it is coal in process of formation from vegetable matter.

Lignite is not found to any extent in Great Britain, but occurs in most European countries, especially in Germany and Austria, and also in America and Australia. Lignites vary, depending on the amount of heat and pressure to which they have been subjected; but their composition in the dry state is, roughly, carbon, 55–80 per cent.; hydrogen, 5–7 per cent.; oxygen, 10–35 per cent.; and generally contain a high percentage of ash and water.

The heating value accordingly also varies considerably, from, say, 5,500 B.Th.U. in inferior earthy lignites to, say, 7,500 B.Th.U. in average good quality lignite. Some American lignites are however, very high in heating value, even up to 12,000 B.Th.U. per lb.

Lignite is used for steam-raising, in making briquettes, and also in distillation processes like coal. By the latter process, it gives an illuminating gas and analogous liquid products to coal.

PEAT

Peat may be said to be a vegetable product intermediate in quality between the original woody matter and lignite (brown coal). Very large deposits occur in various parts of the world, especially in Ireland and various parts of Great Britain. The difficulty of using peat as a fuel is the enormous amount of water it contains, the average figure for natural peat being 80–95 per cent. water, and the difficulty of getting rid of this water. When cut into blocks and dried in the air, although apparently fairly dry, it still contains anything from 30–60 per cent. moisture. A good sample of air-dried peat contains 25–30 per cent. water, with an effective heating value of 7,000–7,250 B.Th.U. per lb. Absolutely dry peat has a heating value of about 10,250 B.Th.U. per lb., and a composition of, say, 58·0 per cent. carbon, 6·0 per cent. hydrogen, 31 per cent. oxygen, and 6·0 per cent. ash. The ash content, however, varies considerably between, say, 1–10 per cent. Air-dried peat is largely used as a household fuel in certain localities, notably in Ireland. The Author recently tested a boiler plant using only air-dried peat as a fuel, and the figures obtained were 2¾ lbs. of water from and at 212° evaporated per lb. of peat. Peat is also used instead of coal in producer plants, and the resulting gas utilized for gas engines. Many processes have been devised to separate the water from peat with a view to using the residue as a fuel, but so far without much success.

WOOD

Wood is used as a fuel for steam-raising on a small scale in countries where large forests exist and coal is scarce. For example, in France, during the last five years, enormous quantities have been used along with coal for steam-raising.

Freshly-cut (green) wood contains a high percentage of moisture, say, 30–50 per cent., depending on the source of the wood. By air drying (seasoning) for, say, 8–12 months, this amount of moisture can be cut down to, say, 20–25 per cent. The same effect can be obtained by drying the wood artificially for a few days at a temperature just over the boiling point of water (212° F.). Such air-dried wood has an effective heating value of about 5,500–6,000 B.Th.U. per lb. Perfectly dry wood contains approximately 50·0 per cent. carbon, 6·06 per cent. hydrogen, 41·5 per cent. oxygen, 1·00 per cent. nitrogen, 1·75 per cent. ash (percentage of ash is, however, variable); and the heating value is anything from, say, 6,500–10,000 B.Th.U. per lb. Some woods, such as pines (conifers), contain small amounts of turpentine and other complicated organic compounds. The above figures are based on the more common woods, such as birch, beech, chestnut, elm, hickory, oak, poplar, and willow. There is not much difference in quality between individual woods regarded as fuels.

BAGASSE

This is the residue of the sugar-cane after it is compressed and the sugar juice extracted, and it is used as fuel for steam generation in sugar factories. The composition and yield of this bagasse depends to some extent on the amount of crushing to which the cane has been subjected. Roughly, 100 tons of sugar-cane will give 35 tons of bagasse residue with single crushing and 28 tons with double crushing.

The composition of the residual green bagasse after double or triple crushing is roughly, say, 50 per cent. woody matter, 9·5 per cent. sugar and other organic substances, 0·5 per cent. ash, and 40 per cent. water. The net calorific value is, say, 2,500–2,750 B.Th.U. per lb. (4,400 B.Th.U. in the dry bagasse) and, under good conditions, 2·25–2·75 lbs. of water from and at 212° can be evaporated per lb. of green bagasse. It burns very well, and a high rate of combustion is obtained.

TANYARD REFUSE (SPENT TAN)

This is the woody residue after the tanning material has been extracted by boiling, and it is produced in large quantities in tanneries. The general practice is to burn it under the steam boilers, chiefly with the idea of getting rid of it. As a rule, only the very poorest results are obtained, because of the enormous amount of water present, which is generally about 65 per cent., although the amount varies. Thus, 100 lbs. of dry bark before extraction gives over 200 lbs. of spent tan. It is an open question as to whether it would pay to compress it in hydraulic presses so as to extract a great portion of the water before burning. The dry tan would have a heating value of about 9,500 B.Th.U.; but because of the large amount of water which has to be evaporated to steam in the fires, the net heating value is only about 2,500 B.Th.U. By careful combustion, this spent tan can be burnt so as to use economically this available heat, and results like 1·75–2·0 lbs. of water from and at 212° per lb. of spent tan can be obtained.

The composition of the dry tan is approximately 51·8 per cent. carbon, 6·0 per cent. hydrogen, 40·75 per cent. oxygen, and 1·5 per cent. ash.

REFUSE

Refuse of almost every description can be economically burnt in a refuse destructor, and the heat produced utilized to heat boiler-feed water, or even generate steam direct in boilers. The heating value of refuse naturally depends on its composition, and is very variable. Such material as straw, paper, and general vegetable refuse will have a heating value of very roughly, say, 2,500 B.Th.U. per lb. General factory refuse can be taken as approximately 2,000–3,000 B.Th.U. per lb., and at least 60 per cent. is combustible. Mixed towns' refuse can be taken as 1,200–2,000 B.Th.U. per lb., and 1–1½ lbs. of water from and at 212° can be evaporated per lb. of refuse.

With the addition of a small amount of coal or coke of inferior quality, the most mixed assortment of refuse can be burnt economically without any trouble, and is really a valuable fuel.

LIQUID FUELS

OIL FUEL (PETROLEUM)

Petroleum occurs as a natural product in the earth, being a thick black or brown liquid accompanied by gaseous products and also solids. It consists of a complicated mixture of hydrocarbon compounds. In organic chemistry, hydrocarbons are divided into two main groups, namely, the "chain" compounds (aliphatic) and the "closed ring" compounds (aromatic), according to the arrangement of the carbon and hydrogen atoms.

The first and typical aliphatic hydrocarbon is methane or marsh gas CH_4, and a series of these aliphatic hydrocarbons is known of the general formulae $C_nH_{2n} + 2$. The first few members are gases at ordinary temperature and pressure (Natural Gas, p. 607); but as their complexity increases, they soon become liquids and, finally, solids. The first and typical aromatic or "closed ring" hydrocarbon is benzene (benzol) C_6H_6, which at ordinary atmospheric temperature and pressure is a liquid, although with a low boiling point (176° F.). As the complexity of this series increased, the boiling point of the liquids also increases until the higher members are reached, which are solids. These aromatic compounds also include a peculiar modification containing more hydrogen, known as the hydromatic compounds. Thus the first of these hydro-aromatic compounds is a liquid hexamethylene C_6H_{12}, corresponding to benzene C_6H_6, the aromatic compounds.

Crude petroleum consists essentially of a mixture of aliphatic, aromatic, and hydro-aromatic compounds, gases, liquids, and solids, although the proportion of liquids is much greater than the gases and solids.

Petroleums from different sources have somewhat different compositions. Thus Pennsylvania petroleum consists chiefly of aliphatic hydrocarbons, and Russian petroleums of hydro-aromatic compounds. Mexican and South African petroleums contain also complicated organic sulphur compounds known as asphaltimes. Of the world's oil production, the British Empire produces only 2·2 per cent. and U.S.A. about 65 per cent. The present consumption of U.S.A. is in excess of production.

The crude petroleum is distilled and separated into various fractions, according to the boiling points. These fractions are then further treated chemically, such as by washing with strong acids or alkalis.

The chief fractions of the distillation of crude petroleum are—

1. *Benzine.*	Consisting of benzine and various amounts of allied compounds. S.G., say, 0·700–0·760 and B.P. 70–150° C.
2. *Paraffin Oil* (*Kerosene*).	A complicated mixture of hydrocarbons, with a S.G. of, say, 0·800 and B.P. 150–300° C.
3. *Gas Oils.*	A further mixture of S.G. 0·850 and B.P. 250° C. and higher.
4. *Fuel Oil.*	Thick liquid residue of 0·880–0·950 S.G. and very high boiling point.

The first product, benzine, etc., is used for the preparation of petrol. Very roughly, petrol is a mixture of hydrocarbons, containing 85–86 per cent. C. and 14–14·75 per cent. hydrogen, corresponding approximately to hydrocarbons: hexane C_6H_{14}, heptane C_7H_{16}, octane C_8H_{18}, of S.G., say, 0·700–0·760 at 60° F., of which about 87–88 per cent. distills over at 120° C. There is always some variation. Petrol has a calorific value of approximately 18,500 B.Th.U. net.

The "kerosene" distillate is generally known as paraffin oil or common lamp oil. Generally has a specific gravity of about 0·800, and is slightly lower in calorific value than petrol. Composition is roughly 87 per cent. C. and 13 per cent. H. It is also used as a fuel for oil engines. Has a calorific value of approximately 18,500 B.Th.U. net.

The higher distillates find a variety of uses. The so-called "gas oil" is used to enrich water gas by vapourizing it and adding to the gas. Diesel oil is a portion of the distillate of about 0·850 S.G. and 18,000 B.Th.U. A third black residue, fuel oil, is used for burning under boilers and also for Diesel engines. Has a S.G. 0·870–0·920.

A considerable amount of oil fuel is now used for steam generation, and within recent years many claims have been advanced for oil burning as against coal.

In burning oil, it is necessary to use "burners," because the oil has to be atomized in order that the air shall mix with it thoroughly to obtain complete combustion. The burners used are of the following general types—

1. *Air-Jet Systems,* in which a mixture of oil and air is blown out of the nozzle by a fan or other means. Generally about 15 lbs. of air per 1 lb. of oil (say, 200 cub. ft.) is used.

2. *Steam-Jet Systems,* using a mixture of steam and oil, in which the oil is atomised by a steam jet blowing in the flame.

3. *Pressure-Jet System.* The oil is drawn from storage tanks by means of a steam-driven pump, and forced out of the nozzles after passing through a strainer and a heater. Heating the oil in this way is claimed to be a big advantage.

As showing the general properties of such oil fuel, the following specifications of the British Admiralty and the United States Government will be of interest.

1. BRITISH ADMIRALTY.

Shall be shale oil petroleum, or some product of petroleum. Flash point not lower than 175° F. close test (Abel or Pensky-Martin flash-point apparatus). Shall not contain more than 3·0 per cent. sulphur and less than 0·05 per cent. free acid calculated as oleic acid. Also less than 0·5 per cent. water. Viscosity on Sir Boverton Redwood's standard viscometer shall not exceed 2,000 secs. for 50 cc. at 32° F. Also shall contain no solid matter liable to choke the burners.

2. UNITED STATES GOVERNMENT.

Flash point not less than 140° F. in closed apparatus (Pemsky-Martin or Abel-Pensky). Specific gravity, 0·85–0·96 at 59° F., and should not congeal or go sluggish at 32° F. Calorific value of not less than 18,000 B.Th.U. per lb. also must not contain more than 2 per cent. water, 1 per cent. sulphur, and any trace of sand or dirt.

There is a very considerable difference of opinion among engineers as to the advantages or otherwise of oil firing, and the main points can be summarized as follows—

It is contended that oil firing is more efficient than coal-firing. There is no great amount of data available on this point, and much further information is required. The Author is of the opinion that 82½ per cent. net working efficiency for complete boiler plant oil fired on the best lines is a generous figure. In comparison with this, a coal-fired boiler plant under average working conditions would give 75 per cent. net working efficiency, so that in this respect there is an advantage with oil.

The heating value of average oil fuel is considerably higher than coal. There is some difference in the heating value of various qualities of oil fuel, but a fair average figure for comparison would be 18,000 B.Th.U. per lb. for oil and 12,000 B.Th.U. for coal. Oil can, however, be very easily contaminated with water, with a serious drop in the heating value.

The great disadvantage of oil fuel is its high price. It is difficult to give average figures, but very roughly oil costs at least £5 per ton, as compared with coal at, say, £2 per ton (1921). On these figures there would be a heavy loss on oil firing after allowing for higher efficiency and better thermal quality of oil. Each case has to be decided on its merits.

As regards the question of stowage space, oil occupies less room than coal; but, on the other hand, requires special facilities in the way of tanks or boilers, which are expensive to install.

Oil is more easily handled than coal, and there is no ash trouble, with a cleaner firehole, and less tendency to smoke.

As regards cost of labour in the firehole, oil-firing reduces labour when the boiler plant is large. For small boiler plants, there is little or no advantage in this respect.

Oil-firing is more dangerous than coal-firing, and there is the risk of explosions in the flues, and also causes greater expense in upkeep of firebrick linings.

An advantage is claimed for oil-firing as an assistant only to coal-firing, using, say, about 10 per cent. of oil compared with coal. This is said to increase the furnace temperature and give good results with poor coal.

SHALE OIL

This is obtained by the destructive distillation of shale, which yields chiefly shale oil and ammonia. The crude shale oil is then fractionally distilled. It is very similar to petroleum, and contains a proportion of what are known to the organic chemist as "unsaturated" hydrocarbons, that is, hydrocarbons containing a less proportion of hydrogen than the ordinary hydrocarbons. The S.G. is roughly 0·945, with a net calorific value of 16,650 B.Th.U. (9,250 calories).

COAL TAR (TAR OIL)

Coal tar is one of the products obtained by the destructive distillation of coal in gasworks and coke-oven plants, being deposited from the gas on cooling. The chemical and physical properties of coal-tar depend to some extent on the coal from which it originates and the method of the distillation. Approximately, however, it is a thin black liquid of S.G., say, 1·05–1·25 and calorific value of, say, 14,850–16,650 B.Th.U. net (8,250–9,250 calories), and consists of an extraordinarily complicated mixture of aromatic organic compounds.

Coal-tar is sometimes used direct as a fuel for steam generation; but what is known as "tar oil" is generally not coal-tar itself, but the residue after some of the more volatile constituents have been separated by distillation. The ordinary crude coal-tar is fractionally distilled at the tar works and separated into various groups of constituents depending on the boiling point. Thus the main groups are benzene, toluol, phenol (carbolic acid), naphthalene, creosote oil, and anthracene. "Tar oil" may, therefore, be a fuel of somewhat varied composition and qualities depending on the amount of distillation to which it has been subjected.

GASEOUS FUEL

BLAST FURNACE GAS

The gases from blast furnaces are a valuable source of heat, both for gas engines and as a fuel for steam-raising. The amount of gas produced is enormous, being about 140,000–180,000 cub. ft. per ton of coke used in the blast furnace, that is, 80–100 cub. ft. from every lb. of fuel used.

An average sample of blast furnace gas is 24¾ per cent. CO, 5¾ per cent. CO_2, 2½ per cent. hydrogen, ¾ per cent. CH_4, and 66½ per cent. nitrogen, with a calorific value of 90–110 B.Th.U. per cub. ft., but the composition varies considerably. It is much better that the gas should be well cleaned before use, even for boiler-firing in special cleaning plants with filters and water sprays, to get rid of dust and tar.

With proper burners, the gas burns very well in boilers, especially of the water-tube type. The evaporation per boiler is reduced, being about 50–60 per cent. as compared with average coal. As regards the efficiency of boilers fired with blast furnace gas, there is not much data published, but with water-tube boilers the figure appears to be from 50–60 per cent. under average conditions.

PRODUCER GAS

Producer gas is sometimes used as a fuel for steam-raising. It is obtained from a number of sources, including anthracite coal, coke, ordinary steam coal, sawdust and wood refuse, lignite, peat, and so on. With anthracite coal, the gas consists roughly of 15–20 per cent. hydrogen, 1¼–1¾ per cent. methane, 22–24 per cent. CO, 48–55 per cent. nitrogen, and 6–6½ per cent. CO_2. The net calorific value per cubic foot is about 150 B.Th.U. With coke, the gas is somewhat poorer, containing, say, 13·0 per cent. hydrogen, 0·5 per cent. methane, 25½ per cent. CO, 55½ per cent. N_2, 5·5 per cent. CO_2, with about 130 B.Th.U. net per cub. ft.

MOND GAS

Mond gas is a variation of producer gas, in which the producer is worked at a lower temperature because of the large amount of steam used. For every ton of average coal, about 2½ tons of steam and 3 tons of air are used in the producer. The products yielded are about 130–150,000 cub. ft. of Mond gas (over 4 tons) and a large yield of sulphate of ammonia (50–90 lbs.). There is not much tar.

The composition of average Mond gas is approximately 20–25 per cent. hydrogen, 1½–3 per cent. methane (CH_4), 13–16 per cent. carbon monoxide (CO), 45–50 per cent. nitrogen, 8–13 per cent. carbon dioxide (CO_2), with a calorific value of, say, 140–160 B.Th.U. per cub. ft. at 60° F. The composition varies, depending on the coal used and the control of the plant.

WATER GAS

Water gas is another variation of producer gas, the method of working the producer being different. The fuel (coal, coke, etc.) is first raised to a white heat with an air blast, and then this is shut off and steam only admitted, this cycle of operations being repeated continuously.

Roughly, the chemical reaction is that the steam (water) is broken up into its constituents hydrogen and oxygen. The hydrogen remains free, and the carbon of the fuel unites with the oxygen to give CO, so that essentially water gas is a mixture of hydrogen and carbon monoxide.

The composition of the gas varies somewhat, but the approximate average composition is 45–48 per cent. hydrogen and 44–45 per cent. carbon monoxide (CO), with 3½–4 per cent. carbon dioxide (CO_2), 2–3½ per cent. nitrogen, 0·5–2·5 per cent. methane, and 1½ per cent. water.

This water gas is often enriched, especially for illuminating purposes, with vapours from heavy hydrocarbons, and is then known as carburetted water gas. The average composition of such enriched gas is approximately 30·0 per cent. hydrogen, 33 per cent. carbon monoxide (CO), 20·0 per cent. methane (CH_4), 11·5 per cent. heavy hydrocarbons, and 5½ per cent. nitrogen, with a heating value of, say, 300–350 B.Th.U. per cub. ft.

COKE-OVEN GAS

Coke-oven gas is largely used for steam generation, being produced in large quantities as a by-product from coke-oven plants in the process of coking coal. The approximate average composition is 50·0 per cent. hydrogen, 25·0 per cent. methane (CH_4), 5·0 per cent. carbon monoxide (CO), traces of carbon dioxide (CO_2), and, say, 20 per cent. nitrogen. The net calorific value is approximately 350–450 B.Th.U. per cub. ft. at 60° F.

COAL GAS

Coal gas is sometimes used as a fuel for steam generation, and gives very good results in the few cases where price is satisfactory. In producing coal-gas from coal in gasworks, the object is to produce as much gas as possible ; and the higher the temperature of distillation in the retorts, the more gas is produced, and the less of other by-products, especially tar. Thus—

Temp. of Distillation (approx.)	Cubic Feet of Gas.	Tar : Gallons.
1800° F.	12,500	8½
1650° F.	11,000	9
1475° F.	10,000	12
1300° F.	9,000	15
1100° F.	7,750	18

The temperature of the distillation effects somewhat the quality of the gas, but under average conditions (1 ton of average coal producing 10–12,000 cub. ft. per ton) the heating value of the gas is 540–580 B.Th.U. per cub. ft. Under these conditions, 1 ton of coal will also produce 8–12 galls. of coal-tar, 20–25 lbs. of sulphate of ammonia, and about 1,344 lbs. (60 per cent.) of coke.

NATURAL GAS

In the United States of America, a fair amount of natural gas is used as a fuel. This natural gas issues from the oil wells at high pressure, and consists chiefly of methane or marsh gas (CH_4) with a complicated mixture of other hydrocarbons. Some of this natural gas contains as much as 96 per cent. of methane. The net calorific value of average natural gas is about 750–1,000 B.Th.U. per cub. ft. at 60° F. and atmospheric pressure. Roughly this natural gas consists of—

Methane (CH_4)	92·5 –94·0 %
Hydrogen..	1·25– 2·5 %
Olefient gas (C_2H_4) (ethylene)	0·15– 0·50%
Carbon monoxide (CO)	0·40– 0·75%
„ dioxide (CO_2)..	0·20– 0·35%
Oxygen	0·30– 0·55%
Nitrogen	2·75– 4·00%
Sulphuretted hydrogen (H_2S)	0·15– 0·20%

J. M. Whitham (*Transactions of the American Society of Mechanical Engineers*, 1905) has tested a number of boilers using natural gas as a fuel. He states that an efficiency of 72–75 per cent. is not usually obtained even with expert attendance, and there is little difference in efficiency with the various burners.

WASTE HEAT

In many industries there is a considerable amount of waste heat which is extremely valuable as a "fuel" for steam generation. For example, coke ovens (non-regenerative), re-heating furnaces, "soaking" pits, puddling furnaces, glass furnaces, various special chemical furnaces, brewers' "coppers," and so on. Most boiler equipment using waste heat in this way is generally in a very poor condition as regards efficiency.

FUEL ANALYSIS.

HEATING VALUE.

From an engineering point of view, the main object of fuel analysis is to determine the heating value of the fuel, that is how many British Thermal Units of Heat per lb. (or other unit) are available for the generation of steam.

The British Unit of Heat is the amount of heat required to heat 1 lb. of water through 1° F. or, more accurately, from 32° F. to 33° F., since there is a very slight difference in the amount of heat required to raise water 1° F. depending on the temperature of the water. The Centigrade or scientific unit is the amount of heat required to heat 1 gramme of water through 1°C. (from 0° C. to 1° C.).

To convert British Thermal Units of Heat to Centigrade Units of Heat, multiply by 5 and divide by 9 and vice versa. The heating value of a fuel is determined with an instrument known as a calorimeter (heat measurer), and in describing methods of fuel analysis, we will primarily consider coal, which is the chief fuel in use.

SAMPLING.

In analysing a fuel, it is absolutely essential to get a proper average sample, and much coal and other fuel analysis is rendered of little value, because of poor sampling. It is impossible to take too large a sample from the whole mass of the coal, and as nearly as possible the large sample should contain the same proportion of large and small, etc., as the bulk. At least, 15–20 shovelsful, for example, should be taken from a truck of coal. The large sample as taken should be then thoroughly broken up with a hammer, so that no pieces are present larger than, say, a small nut. This large sample is then thoroughly mixed with shovels and spread out flat on a clean surface. It is then "quartered," that is divided into four parts, and one of the four parts is taken as the sample. This is then mixed and quartered again, and so on until 4 or 5 lbs. are finally left and put in a sealed tin for analysis. If wooden boxes or bags, or similar receptacles, are used, the sample loses water in transit.

Coal sampling is really a matter of care and common sense and it is difficult to lay down any fixed rule. The Author recommends that two independent samples should be taken and analysed separately, and the results averaged. If the results are much different, a third sample should be taken (*see* also Boiler-testing, page 772).

If the coal is wet, the above methods cannot obviously be used, as the sample rapidly loses water in handling. The same general remarks apply to all fuels—solid, liquid, and gaseous. Coke is particularly difficult to sample, because different portions may contain different amounts of moisture, whilst liquid fuels are easy to sample.

TYPES OF CALORIMETER.

There are many types of calorimeter on the market. Generally cheap instruments are not reliable, as some of the sample is not completely burnt in the calorimeter, or some of the products of combustion may escape at too high a temperature, and secondary reactions due to the chemicals used may generate varying amounts of heat which render the result inaccurate.

The standard type of instrument, both the most accurate and the most convenient to use, is the oxygen bomb calorimeter, where the error is not more than about 0·3 per cent. This instrument consists essentially of a heavy gunmetal bomb, with lid secured by a screw joint or nuts and bolts, and tested to stand a very high pressure, say, 2,500 lbs. per sq. in. The inside of the bomb is lined with some material to withstand the slight corrosive action of the products of combustion. The lid contains underneath a support for a crucible and two electrodes. About 1 gram of very finely powdered coal obtained by roughly grinding up the contents of the sample tin, and then finely grinding an average sample, is weighed out in the crucible on a fine chemical balance and placed in the calorimeter. The ends of the electrodes are connected by a fine platinum wire, which dips in the coal in the crucible. If platinum wire cannot be obtained, a standard length of fine iron wire is used, and a correction made for the heat generated by the combustion of this iron wire to oxide of iron. The bomb is then screwed up, and oxygen from a cylinder admitted through a valve in the lid of the bomb until the pressure in the bomb is 400–500 lbs. per sq. in. The bomb is then placed bodily in the outer vessel of the calorimeter equipment, and a weighed amount of water added, say, 1,500 grams. This water covers the bomb entirely, and the cylinder is air jacketed to prevent any loss of heat. This water is then stirred by a mechanical agitator and the temperature read off on a very accurate thermometer provided to $\frac{1}{100}$° F. When this temperature is recorded, the coal is fired by connecting the electrodes of the bomb lid with several dry cells. The passage of the current heats the wire dipping in the coal white hot, and since the coal is in high-pressure oxygen, the complete ignition is instantaneous. The heat generated heats up the bomb and the surrounding water, the water is stirred as before, and the temperature again read off on the thermometer. It is then an easy matter to calculate the British thermal unit of heat in 1 lb. of the coal, since we know the weights of the coal burnt, the weight of the water, and the rise in temperature.

WATER EQUIVALENT OF CALORIMETER.

It will be obvious that in a calorimeter the heat generated by the combustion of the coal not only raises the temperature of the water, but also the whole body of the calorimeter bomb or other vessel, the stirrer, the thermometer, and other accessories. The heat absorbed in this way is expressed as the water equivalent, that is the weight of water to which this heat corresponds.

The water equivalent of each instrument will naturally be a little different, depending on the construction, and each instrument has to be standardized by burning in it a known weight of some substance of definite heat value and finding the missing "water equivalent" by calculation. The following substances can be used for this purpose—

Cane-sugar	3955·2	calories
Salycylic acid	52·692	,,
Benzoic acid	6322·1	,,
Camphor	9291·6	,,

Once determined, the water equivalent can be stamped on the calorimeter.

FUELS OTHER THAN COAL.

The sample general method of using a calorimeter applies to all fuels. The heating value of any combustible solid is carried out exactly the same way as coal. With liquid fuels, it is generally better to mix a known weight of the fuel in the crucible with some incombustible substance like sand, and with the bomb calorimeter for gases a known volume at a given temperature and at atmospheric pressure is taken. For the minor modifications necessary, the makers' catalogues must be consulted. For gases also, special types of continuous recording calorimeters can be obtained.

GROSS AND NET HEATING VALUE OF A FUEL.

Great confusion is caused by the gross and net heating value of any given fuel, and no definite authority exists on the subject. When an ordinary "dry" coal containing, say, 2½–7½ per cent. of water is burnt in a boiler furnace, the water in the coal is evaporated to steam and presumably escapes to the chimney base in the flue gases at a temperature which may vary from, say, 350°–750° F., depending on the circumstances; also the hydrogen in the coal burns to water, which escapes in the same way. In a bomb calorimeter, this heat does not escape, but the steam is cooled down to, say, 40° F. by the cooling water of the calorimeter. Therefore the B.Th.U. as given by the calorimeter result is slightly higher than in actual furnace conditions and, theoretically, correction ought to be made for this escape of heat from the furnace. The gross heating value is the actual result obtained by the calorimeter, and the net heating value is the gross heating value from which has been subtracted a figure obtained by calculation to correct for the above heat loss on the actual boiler plant, generally taking the exit gases at 212° F.

In actual practice, however, with ordinary dry fuel, this is of little importance, since the difference is only, say, 200–400 B.Th.U. on, say, 12–14,000, and it is customary to take the gross heating value. For wet fuels, however, it is necessary to first dry the fuel and to correct for this.

For the further analysis of fuel, more particularly as applies to coal, we have—

ASH.

The percentage of ash in a coal or other fuel is of great importance, as not only does the amount of ash reduce the heating value of the coal per unit of weight, but is also very serious, because of the great reduction in the efficiency of the boiler plant caused by the obstruction of the ash in the furnaces. In determining the percentage of ash in a coal, 2–5 grams of the powdered coal are weighed out and spread out in a flat silica or platinum dish and heated in a muffle furnace until no organic matter is left, as can be told by the colour. The residue is then the percentage of ash and is weighed off direct. The figure so obtained is generally a little less than the true scientific figure, because some of the ash is apt to be decomposed by the heat of the muffle furnace and be lost by volatilization. It must be remembered also that in a boiler furnace the ash always contains some partially burnt fuel, even under good conditions, so that the percentage of ash obtained from a boiler trial using a number of tons of coal is larger than the results of the laboratory test. Ash itself is chiefly composed of silica, alumina, lime, and iron (oxide and sulphide). Roughly, say, 49½ per cent. silica (SiO_2), 38¼ per cent. iron and alumina, 3¼ per cent. lime (CaO), 1½ per cent. magnesia (MgO), 6½ per cent. sulphate (SO_3), 1 per cent. phosphate (P_2O_3).

MOISTURE.

From a scientific point of view, it is a matter of difficulty to determine the true amount of moisture in a coal. In determining moisture in most substances, the usual method is to heat (that is, dry) at 212° F. or slightly over until no more weight is lost, and the loss in weight represents the amount of moisture. If powdered coal is subjected to this treatment, the water is driven off, but more or less volatile matter may also be driven off. Further, the coal at the higher temperature tends to oxidize and become heavier in weight. Again, when weighing moisture, it is very quickly re-absorbed. By very special precautions, such as heating in coal-gas instead of air, and so on, these complications may be avoided to some extent.

In practice, however, it is understood that "percentage of moisture" really means the loss of weight when heated at 212° F. for one hour in an ordinary drying stove.

About 3 or 4 grams of finely powdered coal are weighed out on a watch glass and spread over the surface. This watch glass is then placed for one hour in a steam-jacketed oven (212° F.). Then it is cooled down in a desiccator (that is in a moisture-free atmosphere) and weighed quickly when cold. If the coal is very wet, the above method cannot be used, as when the coal is ground a large amount of the moisture is lost. The best way, then, is to take a very large sample, say, 50 grams, without handling at all, and determine the moisture in this. The sample can then be ground and the moisture re-determined in the ground sample. The two results will then give the total moisture.

VOLATILE MATTER AND FIXED CARBON.

It is sometimes usual to express the quality of a coal as "percentage of volatile matter" and "percentage of fixed carbon." This is a very crude and rule-of-thumb method of analysis, as the figures obtained depend on the method of carrying out the test. The fixed carbon is supposed to be the percentage of coke less the ash, and the "volatile matter" is the percentage driven off by heating less the water. The standard method of carrying out the test is to weigh out 1 gram of powdered coal in a platinum crucible. The lid of the crucible is fitted on, that is air is excluded. The crucible is then heated by a burner flame about 8 ins. long, with the tip touching the crucible. The heating then goes on until all "flaming" stops, that is no more flames and smoke issue from under the lid of the crucible. The crucible is then cooled in a desiccator and weighed. The weight of the residue represents ash and fixed carbon, and the loss in weight, water and volatile matter. The ash and water are determined separately, as already described; and subtraction gives the percentage of fixed carbon and volatile matter.

SULPHUR.

The presence of sulphur in coal is most objectionable, as it is apt to cause bad clinkering in the boiler furnaces, and also to corrode the boilers or economizers due to sulphurous acid generated by the combustion. It chiefly occurs in coal as iron pyrites, and is found generally as 0·5–3·0 per cent. total sulphur. Sulphur exists in coal in two forms, namely, "volatile sulphur" and "fixed sulphur." The "volatile sulphur" is the objection, as this burns to sulphurous acid. "Fixed" sulphur is the sulphur combined chemically as inorganic salts, such as sulphate of soda, which are not decomposed by the heat of the furnace, are not combustible, and are therefore quite harmless. The percentage of sulphur is determined in the laboratory by a somewhat complicated process, which consists essentially in heating the powdered coal with magnesia (MgO) and sodium carbonate (Na_2CO_3) in a closed crucible, when the volatile sulphur forms sodium sulphide (Na_2S), which is oxidized to sulphate by the addition of bromine water. The contents of the crucible are then boiled with hydrochloric acid, and the sulphate precipitated with barium chloride and weighed as barium sulphate, and from this the sulphur is calculated.

FULL CHEMICAL ANALYSIS.

The full chemical analysis of coal, that is the determination of the percentage of carbon, hydrogen, oxygen, nitrogen, sulphur, etc., is a most complicated chemical analysis, which to-day is of little or no importance in connection with steam generation. Before calorimeters were brought to the present stage of development, the only method of determining the heating value of a coal was to calculate it from the complete chemical analysis. The formula used was Dulongs's, which is as follows—

$$\text{Thermal units of heat} = 145 \left\{ C + 4{\cdot}28 \left(H - \frac{O}{8} \right) + 0{\cdot}28\,S \right\}$$

where C = % of carbon.
H = % of hydrogen and the heating value assumed to be diminished by the amount taken to combine with the oxygen to form water.
O = % of sulphur.
S = % of sulphur.

THE GENERAL PROPERTIES OF STEAM

The word "steam" is generally understood to mean steam that contains no free moisture and is at the exact temperature corresponding to the given pressure, that is, to mean "dry saturated steam." This *dry saturated steam* at, say, 100 lbs. gauge pressure has a temperature of 337·9° F. corresponding to the pressure.

Steam may also contain a considerable amount of condensed water suspended in the steam as vapour, and is then known as "*wet steam.*" The amount of moisture in steam is determined by various forms of a special type of calorimeter, and the percentage of actual steam to total steam and moisture is called the *dryness factor*. In practice, this is not very important in most cases.

"*Superheated*" *steam*, as will be described later, is steam heated up by external heat above the natural temperature of the steam corresponding to the pressure.

THE BRITISH THERMAL UNIT.

There is some confusion at the present time as to the exact definition of the British Thermal Unit of Heat. Roughly, it means the amount of heat required to heat 1 lb. of water through 1° F. For a more exact definition, the trouble is that this amount of heat varies very slightly with the initial temperature of the water. Thus it takes a fraction more heat to raise 1 lb. of water at, say, 200° to 201° than it does to heat 1 lb. of water from, say, 32° to 33°. The older definition is the amount of heat required to raise 1 lb. of water at maximum density (39° F.) through 1° F. Another definition is the amount of heat from 62° to 63° F. Marks & Davis suggest the total amount of heat required to heat 1 lb. of water from 32° F. to 212° F. (180°) divided by 180.

Mechanical equivalent of heat, for all practical purposes, can be taken as 778 ft.-lbs., based on the experiments of Rowlands following the classic researches of Joule.

STEAM TABLES.

On pp. 584–588 are the complete saturated steam tables based on the famous researches of Regnault, and corrected by the further work of many modern investigators, such as Barnes, Callender, Griffiths, Holborn and Henning, Knoblauch and Jolly, Linde, Marks and Davis, Martin and Parsons, etc. From this steam table we have—

The *Sensible Heat* of steam at a given pressure is the amount of heat required to heat 1 lb. of water from 32° F. to the corresponding –° F. at which water is converted into steam at the given pressure. As already seen, the British Unit of Heat is the amount of heat required to raise 1 lb. of water from 32° F. to 33° F., that is 1 unit of heat. But the amount of heat required to raise 1 lb. of water at any original temperature higher or lower than 32° F. is not exactly 1 unit. As the original temperature of the water rises, the amount of heat required to heat 1 lb. of water 1° F. *becomes slightly greater.* Thus from the steam table—

Boiler Pressure lbs. per sq. in.	Absolute Pressure lbs. per sq. in.	1 lb. of water raised from 32° F. to temp. given below, corresponding to the absolute steam pressure.	No. of units of heat actually required.	Theoretical units that would have been required if 1 unit of heat was required to heat 1 lb. Water at any temp. through 1° F.
none	14·7	33° F.	1·0	1·0
,,	14·7	212° F.	180·5	180·0
35·3	50·0	280·9° F.	250·3	248·9
85·3	100·0	327·6° F.	298·1	295·6
135·3	150·0	358·2° F.	329·6	326·2
185·3	200·0	381·6° F.	353·8	349·6

Latent heat of steam is the amount of heat required to convert 1 lb. of water at a given temperature into dry saturated steam at the same temperature, that is, it is the heat lost (latent = lost) in the conversion of the liquid into the gas.

The latent heat of steam really consists of two parts, the first is the internal latent heat absorbed by the molecular change from liquid to gas, that is, the actual intrinsic heat in the steam possessed by the conversion. The second is the external latent heat, that is, the heat absorbed by the condensation of water to steam (i.e. in expanding against the air or existing steam in a closed vessel). The second figure is given on the 6th column, and the internal latent heat is the difference between this figure and the total latent heat. As the temperature of the water rises, the latent heat of steam decreases until at a certain temperature (870° C. according to Regnault) the latent heat would be zero. Thus from the steam tables—

Boiler Pressure lbs. per sq. in.	Absolute Pressure lbs. per sq. in.	Temperature of Water ° F.	Latent heat (i.e. the number of units of heat required to convert 1 lb. of Water at the stated temperature into Steam at the same temperature.
none	14·7	212	966·1
35·3	50·0	280·9	917·3
85·3	100·0	327·6	883·8
135·3	150·0	358·2	861·6
185·3	200·0	381·6	844·6

The latent heat of steam at different temperatures was determined experimentally by Regnault in his monumental researches on steam, and further work has since been carried out by numerous investigators.

The latent heat of steam at any temperature can conveniently be obtained from the following formula—

$$L = \frac{1061{\cdot}3 - 0{\cdot}79\,t}{0{\cdot}956 - {\cdot}000146\,t}$$

Total heat of steam is the total heat required to raise the temperature of 1 lb. of water at 32° F. to a given temperature and to convert it to dry saturated steam at this temperature, and given in the 5th column of the steam table. It includes, therefore, the sensible heat, the internal latent heat, and the external latent heat. It can also be determined by the general formula—

$$H = 1082 + 0{\cdot}305\,t$$

t = temperature of evaporation ;

or more accurately—

$$H = 1150{\cdot}3 + 0{\cdot}3745(t - 212) - {\cdot}00055(t - 212)2$$

As seen from the tables, the total heat of steam increases with the absolute pressure, although the latent heat only decreases. Thus taking the same extract from the tables—

Boiler Pressure : lbs. per sq. in.	Absolute Pressure : lbs. per sq. in.	Latent Heat.	Sensible Heat.	Total Heat.
none	14·7	966·1	180·5	1146·6
35·3	50·0	917·3	250·3	1167·6
85·3	100·0	883·8	298·1	1181·9
135·3	150·0	861·6	329·6	1191·2
185·3	200·0	844·6	353·8	1198·2

Total Heat of Superheated Steam. As already stated, superheated steam is steam at a given pressure heated above the temperature of the formation of steam by external heat.

Thus, for example, suppose we take steam at 128 lbs. gauge pressure, 1142·7 lbs. absolute, with a temperature of 459° F. due to superheating. From the steam tables we see that at 142·7 lbs. absolute, the temperature of the boiling point of water at this pressure is 354·5° F. and the total heat is 1190·1 (864·4 latent heat and 325·7 sensible heat) from water at 32° F. Since the temperature of the superheated steam is 459° F., the amount of rise in temperature due to superheating is 459°–354·4° F., the temperature of formation of saturated steam 104·5° F. But the specific heat of steam is much less than water, that is to say it requires much less heat to raise 1 lb. of water 1° F. than it does to raise 1 lb. of steam 1° F. According to Regnault, the figure was 0·48, but many investigations carried on since have shown that this figure is not quite accurate, and the specific heat of steam depends on the pressure of the steam. The most elaborate investigations on this point are probably by Knoblauch and Jakob (Munich) and by Thomas (America).

Knoblauch and Jakob's results are expressed in the following curve, which is very convenient for the purposes of calculation. In order, therefore, to get the total heat of superheated steam, all that is necessary is to follow up the nearest curve, using the absolute pressure and the temperature of the superheated steam. For example, at 142·7 lbs. absolute and 459° F. superheat temperature, the specific heat is 0·56. The units of heat taken for superheating are, therefore, 459° F. – 354·5° F. = 104·5° F. and 104·5° F. × 0·56 = 58·5 units of heat. The total heat of superheated steam at 142·7 lbs. absolute and 459° F. is, therefore, the total heat of saturated steam, namely, 1190·1 plus 58·5 equals 1248·6 units.

FLOW OF STEAM

1. Flow of Steam through Pipes.

The weight of steam (or other fluid) which will flow in one minute through any given pipe with a given head or pressure is approximately as follows, not allowing for friction in the pipe, number of bends, velocity of the steam, and other indeterminate factors—

$$W = 87\sqrt{\frac{D(P_1 - P_2)\,d5}{L\left(1 + \frac{3{\cdot}6}{d}\right)}}$$

W = weight in lbs.
d = diameter in inches
D = density or weight per cubic foot
P_1 = initial pressure
P_2 = pressure at end of pipe
L = length in feet

The following table will be found of use in the design of pipe lines for velocities up to 100 ft.-sec.

Table of Approximate Capacities of Steam Pipes in Lbs. of Steam per Hour (Saturated)

(*Velocity of Steam* = 100 *feet per second*)

Diam. of Pipe.	Pounds per Sq. In. (Gauge) Pressure.						
	50	**75**	**100**	**125**	**150**	**175**	**200**
2″	1,200	1,640	2,060	2,480	2,900	3,310	3,720
2½″	1,890	2,560	3,220	3,880	4,530	5,180	5,820
3″	2,710	3,690	4,640	5,580	6,530	7,450	8,380
3½″	3,690	5,020	6,310	7,600	8,880	10,150	11,400
4″	4,830	6,550	8,250	9,920	11,600	13,250	14,900
5″	7,530	10,200	12,880	15,500	18,100	20,650	23,200
6″	10,850	14,750	18,600	22,350	26,100	29,900	33,500
7″	14,750	20,100	25,300	30,400	35,500	40,600	45,600
8″	19,300	26,200	33,000	39,700	46,500	53,000	59,600
9″	24,400	33,200	41,800	50,300	58,800	67,200	75,400
10″	30,100	41,000	51,500	62,000	72,500	82,800	93,000
11″	36,500	49,500	62,400	75,000	87,800	100,000	112,500
12″	43,400	59,000	74,200	89,200	104,200	119,000	134,000

Steam pipe dimensions are calculated on following velocities—

Saturated Steam. 3″ pipe or under 75 ft. per sec.
over 3″ pipe, 90 ft. per sec.
Superheated Steam. 9″ pipe or under 120 ft. per sec.
over 9″ pipe, 120 ft. per sec.

Flow of Steam through Nozzles or Small Orifice.

Certain approximate formulae can be used to determine the amount of steam passing through a nozzle of given dimensions at a certain pressure to a lower pressure. The amount of steam issuing from a given nozzle under these conditions increases as the pressure behind the nozzle increases until the external pressure becomes only 58 per cent. of the absolute pressure in the boiler (for dry saturated steam). For superheated steam, the figure is about 55–56 per cent. The formulae (when the final pressure is less than 58 per cent. of the original) commonly used are as follows—

Rateau's formula—$W = 3{\cdot}6\,P(16{\cdot}3 - 0{\cdot}96 \log P)$

W = Weight of steam in lbs. per hour
P = Absolute pressure in lbs. per sq. in.

Rankin's or Napier's formula—

$$W = \frac{6p \times a}{7}$$

W = Weight of steam in lbs. per min.
p = Absolute pressure in lbs. per sq. in.
a = Area in sq. ins.

In connection with steam generation, the amount of steam issuing from nozzles is of great importance, because of the considerable number of mechanical stokers using steam jets (*see* p. 671) and hand-fired steam jet furnace bars (*see* p. 687) in use. The object of these jets is, first, to preserve the bars by the cooling action of the steam, and generally also as a cheap system of forced draught. There has hitherto been very little data available as to the amount of steam used by these nozzles. According to makers' statements, the figure is only about 1–2 per cent. of the steam production, although one firm admits 4–5 per cent.

The Author has devoted a considerable amount of attention to this subject, and published in *Engineering* ("Exact Data on the Running of Steam Boiler Plants: No. 3—The Amount of Steam Used by Steam Jets," 16th Jan., 1920) the results of the investigation into the amount of steam taken in actual practice by 11 types of hand-fired furnaces and 8 types of mechanical stoker. The investigations were carried out over a number of years on 130 boiler plants, comprising 437 boilers with a total annual coal bill of 1,000,000 tons per annum. The average figure obtained worked out at 6·6 per cent. of the production of the plant, a very serious item. The tabulated results are as follows—

AVERAGE RESULTS FOR HAND-FIRING (BROWNLIE)
Averages—6·6 per cent. of the Production

Type of Apparatus.	No. of Plants.	No. of Boilers	% of Production.	Total Coal Bill.	Total Coal Bill used by Jets.
			%	Tons per ann.	Tons per ann.
Type A	6	17	7·6	20,200	1,535·2
„ B	4	7	4·5	15,400	693·0
„ C	3	6	7·3	9,234	674·0
„ D	18	60	6·3	137,005	8,631·5
„ E	7	22	8·1	55,450	4,491·4
„ F	2	3	3·2	5,850	187·2
„ G	2	4	5·0	6,600	330·0
„ H	2	6	7·7	14,400	1,108·8
„ I	3	6	4·4	8,525	375·1
„ J	1	2	15·25	4,000	610·0
„ K	6	16	5·9	35,346	2,079·5
	54	149		312,010	20,715·7

AVERAGE RESULTS FOR MECHANICAL FIRING (BROWNLIE)

Averages—6·7 per cent. of the Production

Type of Apparatus.	No. of Plants.	No. of Boilers	% of Production.	Total Coal Bill.	Total Coal Bill used by Jets.
			%	Tons per ann.	Tons per ann.
Type A. Sprinkling	25	73	5·0	140,345	7,017·2
„ B. „	16	45	5·25	95,550	5,016·4
„ C. „	7	23	5·00	30,070	1,503·5
„ A. Coking ..	4	12	2·30	21,050	484·1
„ B. „ ..	1	3	13·80	5,750	793·5
„ C. „ ..	13	66	8·00	221,950	17,756·0
„ D. „ ..	1	3	7·20	4,900	352·8
„ E. „ ..	9	63	7·50	185,250	13,893·7
	76	288		704,865	46,817·2

MATERIALS OF STEAM PIPES.

Copper, even when solid drawn, is not a reliable material for steam pipes and, when brazed, is particularly dangerous. This applies to ordinary saturated steam as well as to superheated steam.

Cast iron can be used when the pressure does not exceed 100 lbs. gauge for saturated steam. Over this the pipes would have to be so thick as to be clumsy, and there is always the risk of a flaw in the casting. For superheated steam with a moderate amount of superheat, say, not over 100° F., the pressure must be lower, say, 80 lbs. For high superheat, cast iron is quite unsuitable and it is really the best practice not to use superheat at all with cast-iron pipes. Cast iron is very suitable for exhaust pipes.

The Board of Trade formula for cast-iron pipes is—

$$T = \frac{P \times D + \frac{1}{4}}{3500}$$

T = Thickness in inches
D = Diameter inside in inches
P = Pressure in lbs. per sq. in.

In smallest pipes, thickness must not be less than $\frac{5}{16}$ in., and test all pipes by hydraulic to double the pressure.

For pressures of over 100 lbs. and for superheated steam wrought iron or mild steel is used, preferably the latter, and tested to three times the working pressure.

For thickness, the Board of Trade Rule is, with a minimum thickness of $\frac{1}{4}$ in., as follows—

$$T = \frac{P \times D}{6000}$$

This allows a tensile stress of about $1\frac{1}{2}$ tons *per square inch*, and the steel used has generally a tenacity of 25–27 tons.

Flanges are welded on as a rule, but can be screwed. Cast steel pipes are expensive, but it is often better to have awkward bends in cast steel.

Generally speaking, steam pipes up to 12 ins. diameter are now usually solid-drawn weldless pipes of tensile strength 21·6–29·6 tons per sq. in., with an elongation of 27 per cent. in a test length of 2 ins. Larger pipes 13″–16″ are made lap-welded with steel 22–26 tons per sq. in. and 25 per cent. elongation in 8 ins.

BRITISH ENGINEERING STANDARD SIZES OF PIPES, FLANGES, BENDS, AND TEES JOINTS.

For high-pressure superheated steam, the best jointing material is a copper corrugated ring embedded in asbestos, or the copper ring alone may be used and painted with red lead paint. For lower pressures, rubber rings and red lead can be used, or special compressed asbestos rings, or red lead paint only. Many "patent" packings are also on the market.

GENERAL LAYOUT OF STEAM PIPES.

In designing a steam pipe range, two very important factors have to be taken into account, namely, allowance for expansion and contraction, and for avoiding the lodging of condensed water in the circuit and danger of "water hammer."

The expansion and contraction in a steam pipe circuit is very considerable. Thus for cast-iron pipes between 32° and 212° F., the expansion is 0·00000618 of the length per 1° F. For wrought iron, the figure is 0·00000656 and for steel 0·000006. For an ordinary pressure of, say, 160 lbs., a steam pipe will expand roughly 4 ins. in every 100 ft., and with superheated steam the figure may be 6 ins. or even over.

All steam pipes should be carried on rollers (or efficient rubbing surface) fitted on the brackets or columns carrying the pipe, so as to allow perfect freedom for expansion and contraction. Main connecting pipes, such as from the boiler junction valve to the steam main, and from the steam main to the engine or turbine, should be of solid drawn steel, with easy bend to allow for expansion.

In a long range, the centre should be anchored so that the expansion and contraction takes place from this central point, instead of allowing the full expansion of the whole length. The expansion is eased by inserting in the circuit expansion bends of a or shape fixed horizontally. They must not be fixed downwards, because of condensed water. They can be fixed upwards, but this prevents the flow of condensed water. Special telescopic expansion pieces, in which a short length of pipe moves horizontally over an interior pipe, may also be used, but these are generally not so satisfactory as the simple bends.

The "lodging" of condensed water is a very serious matter in a badly designed steam pipe circuit. The pipe from the main junction valve of the boiler must go either horizontally or downwards into the main steam pipe and, if necessary, the junction valve must be raised above the boiler to allow of this. This prevents condensed water lodging between the junction valve and the main steam pipe. The whole range shall slightly "fall" to the end of the circuit from the boiler junction valves, so that all condensed water falls back into the boiler or runs to the end of the circuit. At this end should be placed automatic steam traps; and it is a good practice also to have a valve blowing to air, which is opened up when the steam enters the circuit, so that any large amount of condensed water is blown away at once.

STEAM TRAPS.

Two General Classes—

1. *Bucket and float* traps, in which the condensed water collects in a closed vessel with a float, which opens a valve when the condensed water reaches a given level and allows the water to escape until level is reduced, when the float closes the valve again, and soon only water is discharged and not steam.
2. *Expansion* traps in which the steam is prevented from escaping by a valve closed with a spring or other arrangement; and when condensed water collects this immerses the mechanism and, because of the cooler temperature of the hot condensed water as compared with the steam, causes a contraction of the mechanism and opens the valve allowing the water to escape. As soon as this has escaped the higher temperature of the steam causes an immediate expansion and shuts the valve.

The first class are apt to be heavy and clumsy, and more expensive, but work well in practice and are not easily put out of order. The second class are much smaller and neater in appearance, and cheaper; but are very sensitive, and easily get out of order, even a trace of grit being apt to prevent the expansion and contraction arrangement working effectively.

LOSS OF HEAT FROM UNCOVERED STEAM PIPES.

The loss of heat from an uncovered steam pipe depends on the temperature inside and outside of the pipe, the velocity of steam through the pipe, whether the steam is superheated or saturated, and whether the air outside is stagnant or in motion (i.e. draughty).

The loss is, roughly, 2·75–3·50 B.Th.U. per sq. ft. of heating surface of bare pipe per hour for each ° F. of difference between temperature inside and out (i.e. at 150 lbs. pressure the condensation per sq. ft. per hour of bare pipe is approximately 1·0 lbs. or, expressed in another way, loss units of heat per foot length per hour—

2″ pipe	219·0		
4″ „	390·8	8″ pipe	729·8
6″ „	624·1	12″ „	1077·4

(i.e. 50 ft. of 4-in. pipe at 100 lbs. pressure would condense 40 lbs. per hour, say, about 6 tons of coal a year).

Under good conditions, at least 85 per cent. of the loss is prevented by covering a bare pipe with suitable pipe covering, and the same applies to the boiler.

BOILER AND PIPE COVERING.

An ideal boiler and pipe covering, or lagging, has the following properties—

1. Low heat conductivity. 2. Non-inflammable. 3. Not liable to crack, due to expansion and contraction. 4. Mechanically strong, so as to withstand vibrations and accidental blows. 5. Not acted upon by steam and water, or by the constant contact with high temperature of the pipes. 6. Not heavy, because of the great weight which would be added to the pipe circuit. 7. Should not corrode the pipes.

There are various methods of testing the efficacy of laggings. One method is to take a given length of steam pipe, lag with the material to be tested, and pass in dry steam at a given pressure for a given time, say 2 hrs. Then shut off the steam and weigh the water condensed. The amount of this depends, of course, on the amount of cooling and condensation.

Thompson and Dalby carried out experiments, using a wire heated electrically instead of steam, because of the difficulty of obtaining dry steam. The inside of the pipe was maintained at a constant temperature and by measuring the amount of the current, the loss of heat taking place was recorded.

Darling has further improved this method, using a 32 c.p. filament lamp as the internal source of heat, and has carried out a long series of experiments on various thicknesses and compositions of pipe coverings. For example, in a given case, the bare pipe loss was 1,070 B.Th.U. per sq. ft. per hour; temperature of steam, 400° F.; air, 68° F. 1 in. (85 per cent.) magnesia lost 152 B.Th.U. per sq. ft. insulated pipe per hour, 1½ ins. lost 120 B.Th.U., and 2 ins. lost 100 B.Th.U. After a certain thickness the extra gain is not worth the money.

There are very many substances and compositions in use for boiler and pipe covering, and it is impossible to describe them all. For a high-class job, the best materials are undoubtedly magnesia, slag wool, or fossil meal and asbestos. The boilers should be covered on the same general lines as the steam pipes; and in the case of cylindrical boilers, the fronts should also be covered down to the grate level.

According to Darling—

TABLE OF INSULATING VALUE OF LAGGINGS OF 1½ INS. THICKNESS

B.Th.U. escaping per sq. ft. surface per hour. Steam inside at 400° F. and air outside at 68° F.

85% magnesia	118
Asbestos (good)	121
Mica (best)	123
Inferior asbestos and magnesia mixtures	126
Various plastic compositions	133
Cheap plastic composition	143
Bare pipe	1,070

Magnesia.

This is undeniably one of the finest coverings, especially for high temperatures not over 700° F. Is an almost perfect non-conductor, is very light, and there is not the slightest chemical action over many years by constant contact with the hot steam pipe. The finest magnesia composition consists of 85 per cent. pure calcined magnesia (magnesium carbonate $MgCO_3$) and 15 per cent. asbestos fibre to make a better plastic composition to apply to the pipes.

The general method of application is to mix with water and apply to the hot pipe as a plastic mass, which dries on. The thickness of the covering depends on the conditions as regards the pressure of the steam and the cooling conditions outside. A typical high-class job over 100 lbs. pressure, and especially with superheat, is 2 ins. thick. For less than 100 lbs., the thickness may be, say, 1½ ins. or even 1 in., and for exhaust steam pipes, say, ½–1 in. When wired on, the magnesia should be covered with, say, ½ in. of some hard setting composition to bind the magnesia and make it secure against rough treatment, and then finished off with two coats of paint. For pipes in the open air, a further covering with polished steel sheeting, say, 20_S W.g. thickness is advisable. Also all flanges should be covered with metal flange boxes filled with the magnesia. Magnesia can also be applied in the sectional form in the cases of pipes with considerable vibration.

Slag Wool.

This is also an extremely high-class covering, and is suitable for the very highest temperatures, even over 1000° F. Slag wool is made by blowing steam through molten slag, and consists of a "wool" of very fine fibres of silicates of calcium and iron. It is an extremely good non-conductor, easily applied, and is not effected by years of contact with the steam pipe. In some cases, due to inferior manufacture, prolonged contact with the pipe has generated acid, but with reputable make this is impossible. With this covering an entire metal covering, say, 20-W.g. is necessary to confine the slag wool to the hot surface, but when once applied in this manner the job is permanent. Very suitable for covering boiler fronts and for pipes subject to vibration. A very good job is ½ in. 85 per cent. magnesia on the pipe, with the rest of the covering slag wool.

Asbestos Compositions.

This is a covering of a high quality, consisting of pure asbestos fibre mixed with kieselguhr (fossil meal), which is a deposit of microscopic shells formed of silica, and is non-conducting because of the particles of air enclosed in these shells. It is a very good non-conductor when in a proper fibrous condition and, being little affected by prolonged contact with the hot pipe, it is also mechanically strong. It is applied in the plastic condition or in segments, and further as a rope for very small pipes. Sections can be obtained in various thicknesses from 1–3 inches, covered with painted canvas fastened with metal bands at short intervals. Very good for pipes subject to vibration.

ORDINARY PLASTIC COMPOSITIONS.

All kinds of substances are used in making these cheaper coverings, such as a large proportion of clay mixed with small amounts of asbestos, kieselguhr (fossil meal), plaster of Paris, and waste vegetable fibre (chiefly cotton and flax), and, in general, such mixtures are in reality poor value even at a lower price.

For boiler coverings only, special fireclay slabs are used, and are so arranged with ribs that there is a non-conducting layer of air between the slab and the boiler. Many of these slag coverings are very efficient, and are also very durable.

STEAM BOILERS

THE "CORNISH" BOILER.

THIS boiler was invented about 1797 by Richard Trevithick (a Cornishman), who had the idea of placing the furnace inside the boiler. It is still used to a considerable extent, especially for small installations, such as heating buildings, laundries, potteries, wood-working shops, and so on. "Cornish" boilers are generally of the standard dimensions shown on page 625.

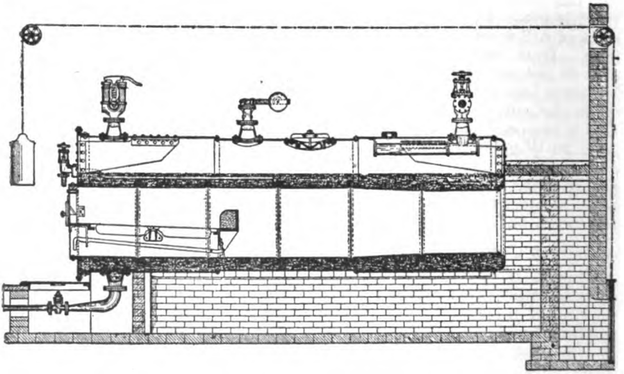

FIG. 1.—CORNISH BOILER.
Longitudinal Section.

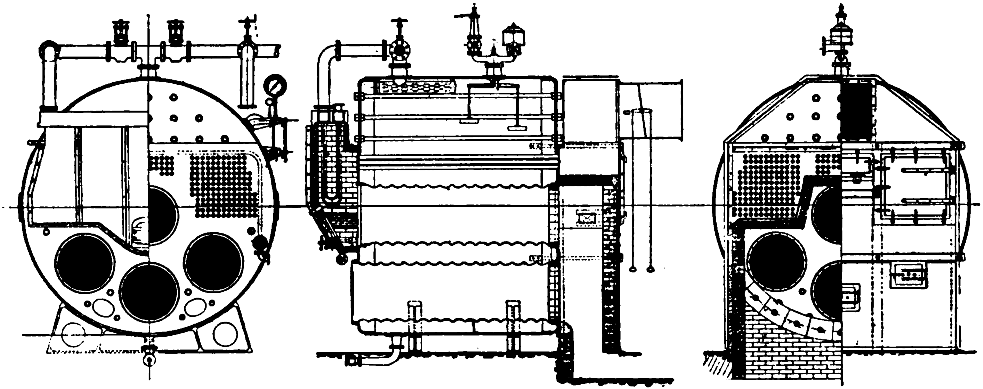

ILLUSTRATION SHOWS LAND TYPE OF "INGLIS" PATENT BOILERS. *For description see pages* 638-639.

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Full particulars on application.

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OF ALL TYPES INCLUDING LOCO TYPE

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"CYCLOPS, GATESHEAD"
"CYCLOPS, FEN., LONDON"

Diameter..	4′ 6″	5′ 0″	5′ 6″	6′ 0″	7′ 0″
Length	15′ 0″	18′ 0″	19′ 0″	20′ 0″	24′ 0″
Approx. wt. boiler & fittings in tons (100lbs.pres.)	4½	5	6½	8	10½
Approx. evaporation per hour in lbs.	1,250	1,750	2,400	2,500	3,250

A typical standard "Cornish" boiler and settings is shown in the illustration supplied by Messrs. Marshall Sons & Co., Ltd., of Gainsborough, suitable for 100 lbs., 125 lbs., or 150 lbs. working pressure.

MARSHALL "CORNISH" MULTITUBULAR BOILER.

As supplied by Messrs. Marshall Sons & Co., Ltd., Gainsborough. The principle of this boiler will be readily understood from the illustration, consisting of a series of small tubes towards the end of the furnace tube, with the object of increasing the heating surface of the ordinary "Cornish" boiler. The makers claim a saving in fuel and increased eva oration.

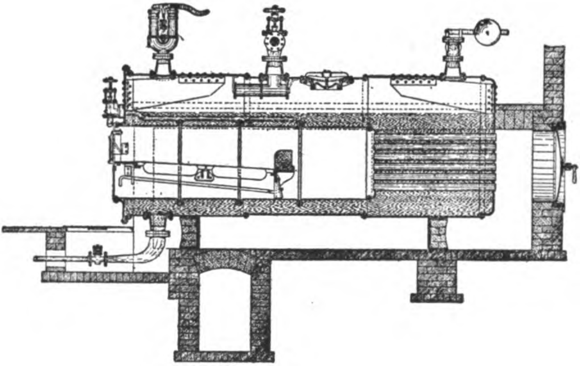

FIG. 2.—CORNISH MULTITUBULAR BOILER.
Longitudinal Section.

PARTICULARS OF "MARSHALL" STANDARD SIZES OF "CORNISH" MULTITUBULAR BOILERS

(*Internally-fired*)

SHELL.		FLUE.		TUBES.		GRATE AREA in Sq. Ft.	TOTAL HEATING SURFACE in Sq.Ft.
Length	Diam.	Length	Diam.	No.	Diam.		
ft. in.	ft. in.	ft. in.	ft. in.		in.		
8 6	3 10	5 10	2 0	16	3	4·6	121
10 0	3 10	5 9	2 0	21	3	6·6	163
10 0	4 5	6 10	2 4	29	3	8·4	204
11 6	4 5	7 0	2 4	34	3	10	243
12 6	4 9	8 0	2 7	37	3	11·6	273
13 9	5 0	9 8	2 9	32	3½	13·1	309
14 3	5 4	9 6	2 9	32	3½	14·5	356
16 3	5 9	11 0	3 0	44	3½	17·5	429
18 6	6 2	12 6	3 3	36	4	21	526
19 0	6 6	12 6	3 6	42	4	24·2	615
21 0	6 9	14 0	3 10	44	4	27·2	707
22 3	7 0	15 6	4 0	52	4	30	800
28 0	7 0	18 2	4 0	52	4	30	1,143

The shells and flues are made of Siemens-Martin steel throughout, of ample strength for working at 100, 125, or 150 lbs. pressure, to meet requirements. The rear end of the flue is continued by a number of best quality fire-tubes from 3 ins. to 4 ins. diameter, according to size of boiler.

The furnace fittings and steam mountings are generally similar to those supplied with their "Cornish" and "Lancashire" boilers.

THE "LANCASHIRE" BOILER.

This well-known type of boiler was invented by Hetherington and Fairbairn in 1841, and first used at Manchester. For this reason it was called the "Lancashire" boiler in opposition to the "Cornish" boiler. It is a minor development of the "Cornish" boiler, having two furnace flues side by side running through the boiler, the diameters of which are reduced towards the end.

The "Lancashire" boiler is made in many sizes from 32′ × 10′ down to as small as 14′ × 5′ 6″.

The following are the general particulars of standard "Lancashire" boilers at standard pressures, as supplied by

Messrs. Yates & Thom, Ltd., Canal Engineering Works, Blackburn, Lancs.—

30′ 0″ × 9′ 0″ *Boiler—*	100 lbs.	120 lbs.	160 lbs.	180 lbs.
Shell plates.. ..	5/8″	21/32″	27/32″	31/32″
Flue rings ..	15/32″	1/2″	19/32″	21/32″
First and last rings of flue ..	19/32″	5/8″	23/32″	25/32″
End plates	11/16″	3/4″	13/16″	13/16″
Stay plates.. ..	19/32″	5/8″	3/4″	7/8″
Shell angle	3½″ × 3½″ × 5/8″	3¾″ × 3¾″ × 3/4″	4½″ × 4½″ × 3/4″	5¾″ × 5¾″ × 7/8″
Weight of boiler and fittings ..	27 tons	29½ tons	35 tons	39½ tons

EVAPORATIVE CAPACITY OF LANCASHIRE BOILERS

Size of Boiler.	Heating Surface.	Grate Area.	*Evaporation per Hour*		
			Rate "A."	Rate "B."	Rate "C."
			Medium Draught 20 lbs.	Good Draught, 25 lbs.	Strong Draught, 30 lbs.
	sq. ft.	sq. ft.	lbs.	lbs.	lbs.
30′ 0″ × 9′ 0″	1,080	44	7,040	8,800	10,560
30′ 0″ × 8′ 6″	1,020	41	6,560	8,200	9,840
30′ 0″ × 8′ 0″	960	38	6,080	7,600	9,120
30′ 0″ × 7′ 6″	900	36	5,760	7,200	8,640
30′ 0″ × 7′ 0″	840	33	5,280	6,600	7,920

"SPECIFICATION OF LANCASHIRE BOILER."

A complete typical specification for a "Lancashire" boiler for 200 lbs. per sq. in. working pressure is as follows—

(For lower pressures, the specification naturally varies, but the general idea is the same.)

SHELL.

The shell to be perfectly cylindrical, and 30 ft. long by 8 ft. diameter, and made from the best selected mild steel boiler plates not less than $\frac{15}{16}$ in. thick; to be in five parallel belts of plating, each being bent to a true circle, and as near equal in width as possible, and *in one entire seamless length.* The joints to be so arranged as to be on the upper part of the shell, and in such a position that when the boiler is seated, they will be clear of the covering of the side flues. All plates to be bent cold. The first ring or belt to be an outer one, to give elasticity to the front end plate. The longitudinal seams to be butt-jointed, with straps inside and outside. The internal straps to be secured by six rows of rivets, and the outside straps by four rows of rivets. The circular seams to be double riveted. The edges of all plates and butt straps to be *planed on their four edges. All the rivet holes to be drilled* by special machinery after the plates are bent and placed in the position they finally occupy, the pitch being accurately defined both longitudinally and circumferentially. *The plates to be taken apart after drilling,* and the *rivet holes slightly countersunk* on both sides to remove the burrs before riveting. A solid angle ring to be riveted to the front shell ring for attaching the front end plate thereto. All rivets, wherever practicable, to be *riveted in by special powerful hydraulic machinery of the most modern construction. All seams to be fullered and caulked inside and outside* after riveting. No plate will be used in the boiler which is not fully up to the specified thickness.

FLUES.

The furnace and flue tubes to be two in number, truly circular, and each constructed of fourteen solid welded rings, 3′–2″ diameter, made from the best selected mild welding steel plates $\frac{5}{8}$ in. thick, and the first and last rings of each flue to be $\frac{3}{4}$ in. thick. Each tube to taper at the last ring but one from the back end to 2′–8″ diameter. The last ring of plating to be parallel. Each ring to be formed by a single plate welded longitudinally and connected by flanges with solid caulking strips placed between. The flanged seams to be arranged so as not to fall in line one with the other or with the ring seams in the shell. Each flange to be formed in one heat by special machinery, and bent to a curve having an inner radius of not less than 1 in., and to have a flat of not less than $2 \cdot \frac{1}{8}$ ins. in width. *The holes in the flues will be drilled by machinery,* specially designed for the purpose to ensure the greatest accuracy, and thus prevent any undue strain being brought to bear on the rivets. *All riveting to be done by hydraulic machinery* of the latest improved principle, *and all the seams to be fullered and caulked on the waterside as well as on the fireside.*

END PLATES.

Each end plate to be in one entire seamless piece to avoid welds and joints, and not less than $\frac{13}{16}$ in. thick and made of the best selected mild steel, *turned on the edges,* and the openings for the flue tubes and mudhole *cut out of solid plate by special machine.* The front end-plate to be securely riveted to a 5¾″ × 5¾″ × ⅞″ external steel angle ring attached to the shell. The back end-plate to be *flanged at one heat in a hydraulic press*, and riveted to the shell ring. The front and back end-plates and shell angle ring to be double riveted.

STAYING.

The ends to be efficiently stayed and strengthened by means of suitable gusset stays, ⅞ in. thick, at each end of the boiler, placed above and below the flue tubes and secured by 4″ × 4″ × ¾″ double steel angles both to the shell and ends, and *designed to give elasticity for the necessary expansion*. The gusset angles and plates will be equal in quality to the shell plates.

MANHOLE.

The manhole mouthpiece to be of wrought steel, of raised circular pattern, 16 ins. internal diameter, of ample strength and size, and provided with a *wrought steel* dished cover, faced, turned, and fitted with the requisite bolts, nuts and suitable lifting ring. The opening through the shell to be cut out by special machine, and strengthened on the inner side by a solid steel hoop. Both the manhole and hoop to be double riveted to the shell, the rivets passing through the three thicknesses.

MUDHOLE.

The mudhole mouthpiece to be of pressed steel, to be oval, and of raised internal pattern and riveted to the front end-plate on the inside of boiler below the flues. To be fitted with a strong steel cover, and wrought iron bolts and crossbars to secure same, together with a neat shell cover for protecting bolts and crossbars. The frame and cover to be accurately faced, to enable the joints being made perfectly tight.

STAND PIPES.

All the requisite stand pipes of pressed steel for the fittings specified to be single riveted to the boiler in the most suitable position, faced on their upper flanges and prepared to receive the fittings to be attached thereto, also one pad block on front end-plate for attachment of the feed valve.

MATERIAL.

The steel used to be of the best quality, made by the "Siemens-Martin" process. To have a tenacity of not less than 26 to 30 tons per sq. in. for the shell and 24 to 28 tons per sq. in. for the flues, with an elongation of about 25 per cent., but not less than 20 per cent., in a length of 10 ins. The rivets to be of mild ductile steel, or good hammered scrap iron of the best quality. The plates to be of one of the following brands. . . . (as specified) . . . , or other improved make.

WORKMANSHIP.

To be of the *very highest class throughout*, and such as will render the boiler and fittings perfectly safe and efficient for a daily working pressure of 200 lbs. on the square inch.

TESTING.

The boiler to be tested before leaving the works by hydraulic pressure up to 300 lbs. per sq. in. and, if required, a certificate of such test having been made will be furnished.

In addition to the usual boiler "mountings" the following are supplied along with the boiler, and included in the specification—

FEED-PIPE.

One perforated cast-iron dispersion feed-pipe for connecting to feed valve, to extend about 20 ft. inside the boiler to effectually distribute the water, the pipe to be carried by suspenders and placed horizontally and parallel with the level of furnace crowns.

BLOW-OFF PIPE.

One cast-steel elbow pipe for connecting the cock to the stand-pipe on boiler of sufficient length to bring the centre of the tap about 12 ins. forward from the front end-plate of boiler.

FURNACE IRONS.

Two furnace fronts and doors with wrought iron frames and neat brass beadings for attaching to the front end of boiler. The doors to be fitted with well-constructed slides to regulate the admission of air for the prevention of smoke, and also provided with internal baffle plates.

FIRE-BARS.

One complete set of chilled furnace bars, with bearers and dead plates for each furnace. The dead plates and the bearers to be carried on wrought iron brackets riveted to the sides of the furnaces.

DAMPERS.

Two sliding dampers and frames for side flues, each with one pulley, stand, wire rope, and balance weight. The frames to be high enough to allow the dampers to rise the full height of the side flues (beams for carrying stands for damper pulleys to be provided). To be geared to front end.

FLUE DOORS.

Two flue frames and doors, machined on joint surface into the brickwork of bottom flue, to give easy access thereto.

FLOOR PLATES.

A complete set of floor plates and frame 8 ft. long for one boiler and continuous if for more than one boiler by 3′–3″ wide, arranged with fender flange for covering the blow-off pit.

ANTI-PRIMING PIPE.

A cast iron anti-priming pipe, perforated on the upper side,

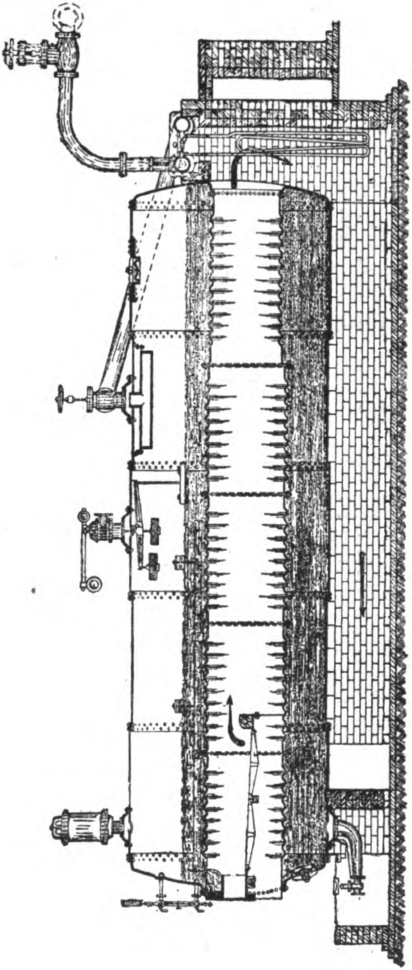

FIG. 3.—DISH-ENDED BOILER.

STANDARD "GALLOWAY" BOILERS

No.	Size. Length × Dia.			No. of Cone Tubes.	Heating Surface.	Normal Grate Area.	Outside Diam. of Furnace.		Steam Evaporated per Hr. lbs.			Approximate Net Weight.	
									Coal burnt per Sq. Ft. Grate per Hour.			Working Pressure.	
									20 lbs.	25 lbs.	30 lbs.	120 lbs.	160 lbs.
	Ft.	Ft. Diam.	Ins.		Sq. Ft.	Sq. Ft.	Ft.	Ins.	Lbs.	Lbs.	Lbs.	Tons.	Tons.
1	14 ×	5	6	6	300	14·5	2	1	2,500	3,100	8,700	7	—
2	16 ×	5	6	8	350	16	2	1	2,700	3,400	4,100	7·5	—
3	18 ×	5	6	8	395	17·2	2	1	2,900	3,600	4,400	8	—
4	16 ×	6	0	8	395	19·5	2	4	3,300	4,100	5,000	8	—
5	18 ×	6	0	11	445	21	2	4	3,500	4,800	5,300	8·75	—
6	20 ×	6	0	11	495	22·5	2	4	3,800	4,800	5,700	9·5	—
7	22 ×	6	0	14	555	22·5	2	4	3,800	4,800	5,700	10·3	—
8	20 ×	6	6	15	570	23·2	2	7	3,900	4,900	5,900	10·5	12·5
9	22 ×	6	6	15	605	25	2	7	4,200	5,300	6,400	11	13·25
10	24 ×	6	6	18	675	27·5	2	7	4,600	5,800	7,000	11·8	14·5
11	26 ×	6	6	18	725	30	2	7	5,100	6,400	7,600	13	15
12	24 ×	7	0	18	735	29·7	2	9	5,100	6,300	7,600	14·5	17
13	26 ×	7	0	20	795	32	2	9	5,400	6,800	8,200	15·5	18
14	28 ×	7	0	20	850	32	2	9	5,400	6,800	8,200	16·5	19
15	30 ×	7	0	25	925	32	2	9	5,400	6,800	8,200	17·5	20
16	28 ×	7	6	20	925	35	3	0	5,900	7,400	8,900	17·75	22
17	30 ×	7	6	25	1,000	35	3	0	5,900	7,400	8,900	19	23·5
18	28 ×	8	0	25	1,070	38	3	3	6,500	8,100	9,700	20	25
19	30 ×	8	0	25	1,120	38	3	3	6,500	8,100	9,700	21	20
20	30 ×	8	6	25	1,220	40	3	5	6,800	8,500	10,200	23·5	28·3
21	30 ×	9	0	25	1,310	43	3	8	7,300	9,100	11,000	25·5	31·5

Basis—8·5 lbs. of water evaporated per lb. of coal. Boiler supplied with hot feed as from a Green's economizer.

to be placed horizontally inside the boiler near the top, and connected with the stand pipe for steam junction valve.

BOLTS.

All the requisite joint bolts for the fittings named.

Various extras can also be included if desired, such as—

I. The front end-plates to be flanged outwards for attachment to the flues.

II. A length of, say, 9′–0″ of the furnace tube to be corrugated towards the back-end.

"DISH-ENDED" "LANCASHIRE" BOILERS—

. . . introduced by *Messrs. Thompsons, of Wolverhampton*, in which the end-plates are "dished" to a convex form instead of the usual flat ends, and the furnace tubes corrugated to allow for contraction and expansion.

This type of boiler (page 631) is now made by a number of firms.

"GALLOWAY" BOILERS—

. . . as supplied by *Messrs. Galloway & Co., Ltd., of Manchester*. This is a "Lancashire" boiler modified by having the two furnace tubes joining into one flue of a special shape—containing the "Galloway" cone tubes, the object of which is to increase the circulation of heater in the boiler. (*See* Table on p. 632.)

"PAXMAN ECONOMIC" BOILER—

. . . as supplied by *Messrs. Davy Paxman & Co., Ltd., Head Office, Colchester.*

This boiler is generally supplied with two furnace tubes, like a "Lancs" boiler, but the boiler is fitted with a number of small tubes fitted horizontally in the top of the boiler, as seen in the illustration.

The makers claim that this boiler gives a very greatly increased heating surface as compared with the "Lancs" and similar types. As a consequence they claim that the temperature of the exit gases is only about 450–475° F. instead of 600–700° F. as in the case of the "Lancs" boiler.

The "Economic" type of boiler consists of a cylindrical shell, with one or more internal flues, and a number of return tubes running from end to end. A combustion chamber of brick receives the products of combustion at the back of the boiler: they then pass through the small tubes to the smokebox in front, and finally return to the back, outside the boiler shell, through the brick flues in the setting, to the chimney. A great advantage of the brick combustion chamber at the back is that this becomes practically red hot when the boiler is at work, thus ensuring the almost perfect combustion of the gases, and reducing the smoke to a minimum.

In the "Paxman Economic" boiler, the tubes (which form some of the most effective heating surface in the boiler) can be swept daily, thus keeping them at their maximum efficiency. The operation of sweeping the tubes can be effected by the stoker, using an ordinary wire brush, whilst the boiler is at work, during any interval of 30 minutes : or, if there is no such interval, a steam-cleaning device can be supplied, enabling the operation to be completed in ten minutes. With boilers of the "Lancs" and similar types, it is impossible to clean any part of the heating surface while the boiler is at work.

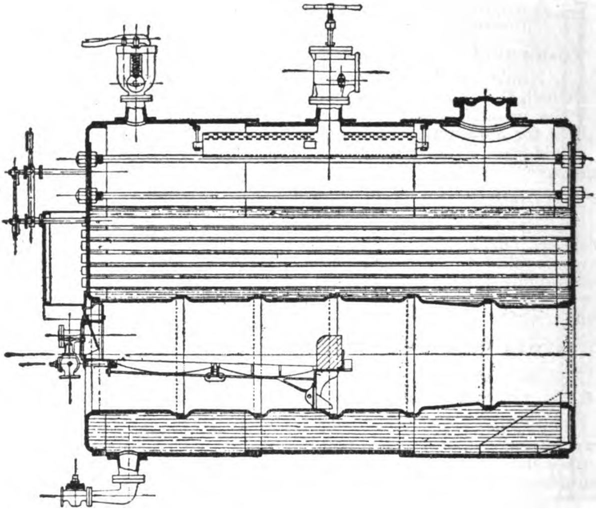

FIG. 4.—PAXMAN DRYBACK BOILER.
Longitudinal Section of Boiler.

It is also stated that the "Paxman" boiler occupies much less floor space than the "Lancs" boiler. It is made for all pressures from 90 lbs. to 180 lbs.

The figures given, p. 635, for the evaporate capacity of each boiler are with feed at 212° F., when burning good coal, and with good draught and stoking. It is, of course, possible to obtain a much greater evaporation when the draught is ample, and the user is prepared to sacrifice economy to increase the evaporation.

PRINCIPAL DIMENSIONS OF STANDARD SIZES

Length of Boiler.		Diam. of Boiler.		No. of Flues.	Diam. of Flues.		Evaporation per Hour.	Approx. Weight of Boiler for a Working Pressure of		Approx. Weight of Fittings.		Approximate Measurement.	
								100 lbs.	120 lbs.	Un-packed.	Packed in Case.	Boiler.	Fittings in Case.
ft.	in.	ft.	in.		ft.	in.	lb.	cwt.	cwt.	cwt.	cwt.	cub. ft.	cub. ft.
6	6	4	9	1	2	3	840	43	45	18	21	210	54
7	0	5	0	1	2	4	1,040	48	50	19	22	252	59
8	0	5	0	1	2	6	1,400	57	59	20	23	285	62
8	6	5	0	1	2	6	1,480	59	61	22	25	303	66
9	0	5	3	1	2	6	1,650	66	72	24	28	350	72
9	6	5	6	1	2	9	1,940	78	84	28	32	420	78
9	6	6	0	1	3	0	2,350	90	95	30	35	495	85
11	0	6	3	1	3	2	2,860	118	130	36	41	580	100
12	6	6	6	1	3	2	3,400	130	155	40	45	710	112
12	6	7	0	2	2	3	3,990	164	176	53	59	807	145
12	6	7	6	2	2	4	4,530	182	198	56	62	920	150
14	0	7	6	2	2	6	5,220	200	220	58	64	990	160
14	0	8	0	2	2	8	6,960	218	242	60	67	1,150	165
14	0	8	7	2	2	10	7,890	265	290	74	80	1,320	—
15	6	8	10	2	3	0	9,100	310	325	78	85	1,464	—
15	6	9	9	2	3	2	11,500	380	410	80	90	1,810	—

THE "HUDSON" COMBINED CYLINDRICAL AND WATER-TUBE BOILER—

. . . as supplied by *Messrs. Thomas Hudson, Ltd., Sheepford Works, Coatbridge, Scotland.*

This boiler is made in two types. For large evaporation, the cylindrical element consists of a short two-flued boiler of the "Lancs" type, but for lesser evaporation. This central boiler is of the "Cornish" type, and is flanked on each side with a nest of water tubes set low down in front, and rising towards the back. The bottom header of nest of tubes is connected by a mild steel pipe of large capacity to a mud-drum at the fore end of the central boiler. Each back tube header is connected to the steam space of the boiler by a mild steel bend pipe of sufficient size to take all the steam generated in each water tube element. The flue gases pass from the central boiler furnaces and sweep along and over each of the water-tube arrangements, and the side of the central boiler, and then pass into the main flue under the central boiler, and on to the main flue, economizer, and chimney, as usual. The hottest gases, therefore, do not come in direct contact with the tubes.

The circulation of water in the boiler is very effective, and is obtained by feeding the water through the front end as usual: the feed water, being colder than the water in the body of the boiler, quickly gravitates to the bottom of the cylindrical element, and thence passes through the connections aforementioned into the bottom drums or headers of the water-tube elements. The water then passes up the tubes, and is rapidly heated as it approaches the hottest gases travelling in a contra-flow direction. As the steam is formed, it quickly makes its way up the inclined tubes to the back header, and then into the steam space on the cylindrical element. Owing to the method of feeding the water-tube elements, there is a definite circulation of water in the cylindrical element of the boiler, which is thus kept at a more uniform temperature than in the case of "Lancs" boilers, and thereby attains a very important object in preventing strains due to unequal expansion and contraction.

THE "YORKSHIRE" BOILER—

. . . as supplied by *Messrs. Holdsworth & Sons, Ltd., Croft Boiler Works, Bradford.*

The "Yorkshire" boiler is constructed on the same general principles as the "Lancashire" boiler, except that it is shorter, its length (max. 24′–0″) being about 2½ times its diameter, and the furnace flues expand from front to rear in the proportion of 2 to 3.

STANDARD HUDSON PATENT BOILERS

List No.	Nominal Evaporation in Lbs. per Hr.	Dimensions.			Heating Surface.	Grate Area.	Ratio H.S./G.A.	Space Occupied.	
		Cylindrical Element.		2 Water Tube Elements each consisting of	Sq. Ft.	Sq. Ft.		Width over Brick-work.	Length over Brick-work.
1	17 000	18′ 0″ × 9′ 0″	Double-Flued	57 tubes 4″ o/d	3,065	44	69·6	21′ 3″ ×	24′ 6″
2	15,000	18′ 0″ × 9′ 0″		45 ,,	2,561	44	58·2	20′ 3″ ×	24′ 6″
3	14,000	18′ 0″ × 8′ 6″		45 ,,	2,489	41	60·7	19′ 9″ ×	24′ 3″
4	13,000	18′ 0″ × 8′ 0″		45 ,,	2,427	38	63·8	19′ 3″ ×	24′ 0″
5	12,500	17′ 0″ × 8′ 0″		45 ,,	2,307	38	60·7	19′ 3″ ×	23′ 0″
6	11,500	17′ 0″ × 7′ 6″		40 ,,	2,080	35	59·4	18′ 0″ ×	23′ 0″
7	11,000	17′ 0″ × 7′ 0″		40 ,,	2,048	32	64	17′ 6″ ×	23′ 0″
8	10,000	17′ 0″ × 7′ 0″	Single-Flued	37 ,,	1,928	32	60·2	16′ 9″ ×	23′ 0″
9	7,500	17′ 0″ × 7′ 0″		25 ,,	1,346	22	61·2	16′ 3″ ×	22′ 6″
10	6,200	17′ 0″ × 6′ 6″		22 ,,	1,194	20·5	58·2	15′ 9″ ×	22′ 6″
11	5,800	16′ 0″ × 6′ 0″		21 ,,	1,075	17·4	61·7	14′ 6″ ×	21′ 3″
12	5,200	16′ 0″ × 5′ 6″		21 ,,	1,030	16	64·3	14′ 0″ ×	21′ 3″
13	4,000	14′ 0″ × 5′ 0″		16 ,,	745	13·3	56	13′6″ ×	19′ 3″

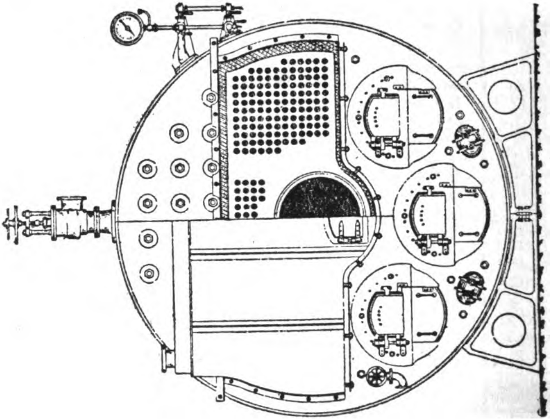

FIG. 6.—INGLIS.

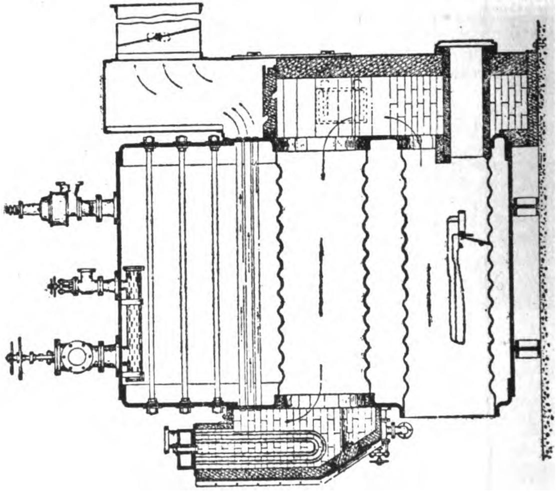

FIG. 5.—INGLIS BOILER.

THE "INGLIS" BOILER—

. . . as supplied by the *Inglis Boiler Syndicate, Ltd.*, 83 *Renfield Street, Glasgow.*

This boiler is a modification of the ordinary Marine boiler, and is made with two, three, or four furnaces—depending on the size of the boiler. The furnaces are all connected at the back of the boiler to a common combustion chamber, which is not part of the shell of the boiler, as seen in the illustration. From this common combustion chamber a return flue is led back through the boiler again to the front-end opening into a brick-lined chamber. This chamber extends over the entrance to all the tubes, then leads through the boiler again to the back smokebox and thence to the chimney.

THE "SPENCER BONECOURT" (KIRKES PATENT) BOILER—

. . . as supplied by *Messrs. The Bonecourt Waste Heat Boiler Co., Ltd., Parliament Mansions, Victoria Street, London, S.W.*1.

This boiler can only be fired with coal in combination with a gas producer. It is more particularly adapted for use with gaseous fuels, with crude oils or paraffins, and waste heat.

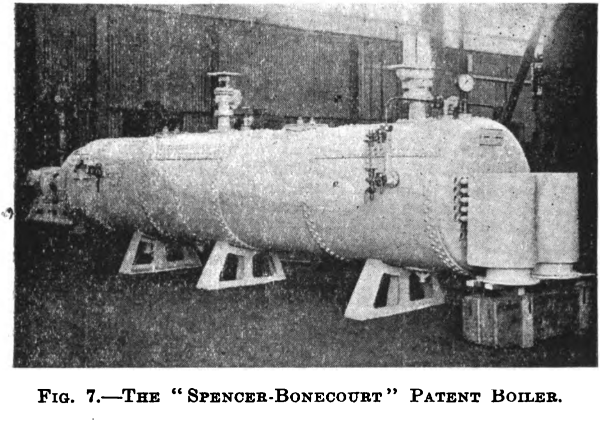

FIG. 7.—THE "SPENCER-BONECOURT" PATENT BOILER.

Advantages claimed for the "Bonecourt" boiler are said to be due to—

1. Shape of combustion chambers. 2. Packing in the tubes.

1. *Combustion Chamber.*—This consists, as seen in the illustration, of a number of separate combustion chambers of a very small cross-sectional area (only a few inches in diameter),

but of considerable length. If, therefore, air and gas are introduced from separate pipes in correct amounts, they are bound to mix intimately because of the small area of the combustion tubes. Consequently, the combustion is almost complete with the theoretical amount of air, and it is claimed that, in practice, only about 10 per cent. excess air is used.

2. *Packing in the Tubes.*—The small combustion chambers, or tubes, are packed for the whole or part of their length with a special iron packing in the shape of a spiral. The object of this special packing is to accelerate the mixing of the air and the fuel, to increase the radiation effect, increase the length of travel to the gases, and to sweep away the inert cold film which adheres to the walls of an ordinary straight tube.

This boiler is generally used with superheater, economizer, and mechanical draught; and the figure shows a typical installation of this nature.

VERTICAL BOILERS

"HUDSON" VERTICAL BOILER—

. . . as supplied by *Messrs. Thos. Hudson, Ltd., Sheepford Works, Coatbridge, Scotland.*

Consisting of two cylindrical elements joined by straight water tubes of one length and placed that they can be easily removed through manhole on top element. The bottom element contains firebox, lined with firebricks, which lining materially assists combustion and gives increased water space between shell and firebox, where no mud sediment can form in the boiler as in water-jacketed types.

The gases from firebox pass through the tapered uptake to the top element and among the water tubes. They then travel downwards between shell and outer casing (which is asbestos lined), and through suitable portholes to the chimney. A large heating surface is provided, giving the boiler a high thermal efficiency.

The two elements are connected by an external circulating pipe, on which the feed valve is placed. The hot water from top element is mixed with the feed, which is raised in temperature.

THE "COCHRAN" VERTICAL BOILER—

. . . as supplied by *Messrs. Cochran & Co., Ltd., Annan, Scotland.*

This boiler does not require any brickwork setting, and all that is necessary is a hard level surface. It has a special arrangement of horizontal flue tubes, which give a large heating surface on the principle of the locomotive boiler. It is also wholly accessible for cleaning and examination inside.

These boilers are all supplied with standard mountings complete, namely, stop valve, safety valve, feed-check valve, blow-off cock, water gauge, test cocks, and pressure gauge.

"COCHRAN" BOILERS.—LIST OF STANDARD SIZES (*English Measure*)

No.	Dia.		Height.		Heating Surface.	Grate Area.	TUBES.		FUNNEL.		EVAPORATION.		APPROX. WEIGHT WITHOUT WATER.		No.
							No.	Ex. Dia.	Dia.	Height Recommended.	Heavy Steaming.	Easy Steaming.	Bare Boiler including Back Plate and Lining, Angle Ring, & Brackets for Firebars.	Complete with Mountings, but no Funnel.	
	Ft.	In.	Ft.	In.	Sq. Ft.	Sq. Ft.		In.	In.	Ft.	Lbs. per hr.	Lbs. per hr.	Cwts.	Cwts.	
1	3	0	6	9	60	4·75	49	1¾	9	11	360	270	23	29	**1**
2	3	3	7	6	80	5·75	54	1¾	10	12	500	375	25	33	**2**
3	3	9	8	6	100	7·50	50	2	11	13	700	540	32	41	**3**
4	4	0	9	0	120	8·50	64	2	12	14	850	651	38	48	**4**
5	4	3	9	6	140	9·25	58	2½	13	15*	1,000	747	45	54	**5**
6	4	6	10	0	160	9·75	65	2½	14	16	1,100	823	50	60	**6**
7	4	9	10	3	200	11·75	74	2½	15	18	1,400	1,050	55	68	**7**
8	5	0	11	3	220	12·50	84	2½	16	20	1,520	1,148	65	78	**8**
9	5	3	11	9	250	14·00	92	2½	17	22	1,760	1,320	73	88	**9**
10	5	6	12	3	300	16·75	102	2½	18	24	2,160	1,620	82	98	**10**
11	5	9	13	0	350	18·75	123	2½	20	26	2,500	1,878	95	114	**11**
12	6	0	12	6	350	18·75	110	2½	20	28	2,500	1,878	96	115	**12**
13	6	0	13	6	350	18·75	110	2½	20	28	2,500	1,878	100	118	**13**
14	6	0	14	0	400	18·75	136	2½	21	28	2,600	2,002	107	127	**14**
15	6	6	13	6	450	22·50	143	2½	24	30	3,100	2,358	116	140	**15**
16	6	6	14	0	450	22·50	143	2½	24	30	3,100	2,358	121	145	**16**
17	6	6	14	6	500	22·50	158	2½	24	30	3,300	2,478	125	148	**17**
18	7	0	14	0	500	26·75	143	2½	25	32	3,600	2,748	140	165	**18**
19	7	0	15	0	600	26·75	173	2½	25	32	4,000	3,015	150	176	**19**
20	7	6	16	3	730	31·50	216	2½	29	34	4,800	3,646	190	228	**20**
21	8	0	16	6	850	37·00	224	2½	31	38	5,700	4,320	220	258	**21**
22	8	6	17	0	1,000	41·00	281	2½	34	40	6,600	4,957	255	304	**22**

Note.—The figures in above table give particulars of our Standard Boilers (100 lbs. W.P.).
Boilers up to 4 ft. 9 ins. dia. inclusive have dished tops. Boilers above this size have hemispherical tops.
Boilers 4 ft. 3 ins. and upwards have funnels fitted with shackles and guy ropes. The smaller funnels are not guyed.

* Funnels over 15 ft. high are supplied in lengths for shipment, no length exceeding 15 ft. Angles are fitted to the sections for bolting together at destination.

HORIZONTAL FLUE TUBES

Height .. .. 14′-0″ Heating surface 400 sq. ft.
Diameter .. .. 6′-0″ Grate surface .. 18·75 sq. ft.

Scale $\frac{5}{16}$″ to 1 foot

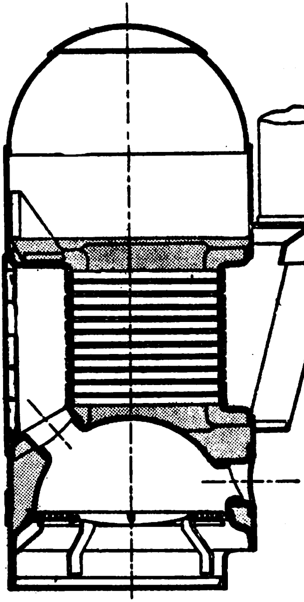

FIG. 8.—7 FT. 6 IN. DIAMETER COCHRAN BOILER.
Sectional View.

WATER-TUBE BOILERS

The water-tube boiler was evolved for various reasons—chiefly a desire for extra efficiency by increasing the speed of the circulation of the water in the boiler. Also for reduced floor space, reduced weight (especially for marine work and conveyance over rough country), and for quick steam-raising and quick repairs for marine work.

The tendency of all boiler construction is to bring the fires into as intimate contact with the water as possible, and this is obtained both by the construction of the water-tube boiler and the increase in the circulation of the water past the heated plates or tubes.

For ordinary industrial land boiler plants, it is not an easy matter to decide which type of boiler is best for the given conditions, and very many factors have to be taken into account. A short analysis of these is as follows—

1. EFFICIENCY.

Generally a water-tube boiler is more efficient than a "cylindrical" boiler. Under good conditions, a water-tube boiler will run on, say, 70 per cent. efficiency, and a cylindrical boiler on, say, 65 per cent. efficiency.

If, however, the requisite number of economizers and super-heaters be added to the boilers, there is little difference. A complete water-tube boiler plant will then run on, say, 77½–80 per cent. efficiency and a cylindrical plant on, say, 75–77½ per cent. under equal conditions.

2. STEAM PRESSURE.

For high pressures, 180–200 lbs. and over water-tube boilers are, on the whole, preferable. In practice, "Lancashire" boilers are better not over 180 lbs., although numbers are at work at 200 lbs.

3. QUICKNESS OF STEAM-RAISING.

The water-tube boiler is enormously superior to the "Lancashire" boiler in this respect. In practice, however, this is not a great advantage, as the average industrial plant does not need to suddenly get up steam.

4. MECHANICAL-FIRING.

It is a great advantage of water-tube boilers that efficient mechanical-firing can be carried out. Mechanical-stoking is much more difficult to apply satisfactorily to "Lancashire" or other cylindrical boilers (*see also* pp. 656–666).

5. FLOOR SPACE.

Very much less floor space for a given steam production is required for water-tube boilers. For moderate-sized plants under average industrial conditions, such an advantage is, however, not very important.

6. WEIGHT OF PLANT (*on foundations*).

Here, again, the advantage is most decidedly with the water-tube boiler, but in most cases is not of much importance.

7. EASE OF TRANSPORT.

A water-tube boiler is obviously much easier to transport, as not only is the total weight for a unit of evaporation much smaller, but it is carried in sections and put together on the site. For many countries this is a great advantage.

8. FEED-WATER.

It is an advantage, in practice, of the "Lancashire" or other cylindrical boiler that it will work much better with moderate or bad quality feed-water. A water-tube boiler, with water of even 10° hardness, is in trouble, the tubes being "made up" and damaged by the heat, and the efficiency of the boiler seriously reduced. The cost of scaling out the boiler is also a big item as against the "Lancashire" boiler.

For the best results, a water-tube boiler requires not over, say, 5–6° total hardness; and whilst good water is a necessity for the best results on any type of boiler with moderate water, the efficiency of a "Lancashire" boiler is not reduced anything like so much as a water-tube boiler.

Grease is also particularly objectionable in water-tube boilers, and the same applies to corrosion.

9. Breakdowns.

A "Lancashire" or similar boiler is much more free from stoppages and breakdowns than a water-tube boiler. It is quite common for "Lancashire" boilers to run ten years without any repairs, and a total of 20–25 years is quite common.

Such boilers made of old Low Moor iron have lasted an extraordinary time, and the writer is acquainted with one plant that has been running for fifty-five years. For smaller industrial concerns, this is a great advantage.

10. Simplicity.

It is a great advantage of a "Lancashire" or other cylindrical boiler that it is simple in construction and simple to work. Unskilled attention in a water-tube boiler can very quickly result in serious trouble.

11. Reserve of Steam.

A "Lancashire" boiler contains an enormous "reserve" of steam, because of the large amount of water contained in it as compared with a water-tube boiler. Consequently it is much easier of a water-tube fluctuating loads. In the case of a water-tube boiler, an automatic feed-water regulator is absolutely necessary under such conditions.

The "Babcock & Wilcox" Boiler—

As supplied by *Messrs. Babcock & Wilcox, Ltd., Oriel House, Farringdon Street, London, E.C.*

This boiler is supplied in four general types, but the best known type for ordinary land installations is the "Class W.I.F." as illustrated. (Fig. 9.)

The boiler consists of three essential parts, namely—

I. The tubes inclined over the furnace, as shown in the illustration. These tubes are made of wrought steel expanded at each end into sectional steel headers, each connecting one zigzag row of tubes. These connecting headers are provided with handholes for cleaning purposes opposite each tube, and the handhole covers have a metal-to-metal joint to do away with the trouble due to joints made with perishable material. The covers are held in place by wrought steel clamps. The connecting boxes, or headers, are connected to the main drum by short tubes expanded into holes in the drum.

II. A horizontal drum for steam and water, constructed of mild steel, double riveted or butt-strapped, and fitted with the usual mountings.

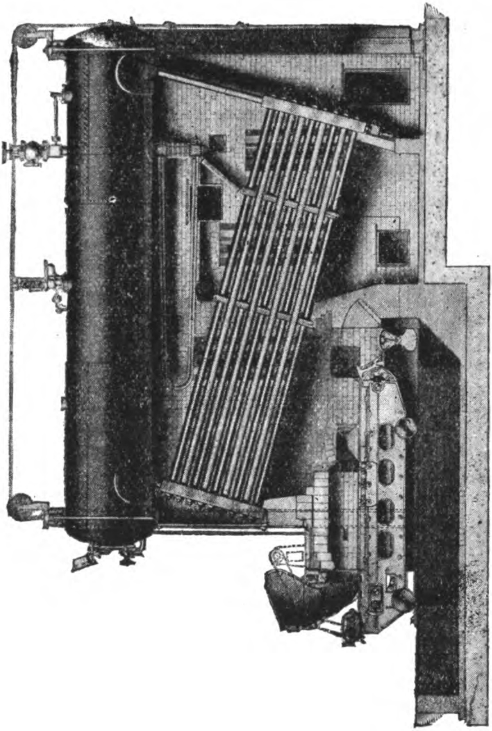

FIG. 9.—BABCOCK & WILCOX BOILER.
Fitted with Integral Superheater and Mechanical Chain Grate Stoker.

III. A mud collector of wrought steel attached to the lowest portion of the inclined tubes in which mud and precipitated matter is collected.

The water circulates through the tubes, rising to the front header and up into the boiler, whilst the return is down the back header, giving a rapid and continuous circulation in one direction. The tubes are "zigzagged" and effectively break up the hot gases as they rise.

The boiler is entirely supported by wrought iron girders and columns, independent of the bricksetting, so that it can expand and contract without undue strain, and is easy of access.

Over 22,300,000 h.p. have been installed or on order throughout the world for land and marine service.

THE "KESTNER" WATER-TUBE BOILER—

. . . as supplied by *Messrs. The Kestner Evaporator and Engineering Co., Ltd.*, 37 *Parliament Street, Westminster, London, S.W.*

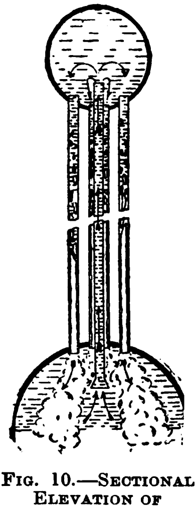

FIG. 10.—SECTIONAL ELEVATION OF BOILER, SHOWING RETURN TUBES.

As seen from the illustrations, the "Kestner" boiler consists of a steam or upper drum, and a mud or lower drum. The two are united by a double row of (3) tubes, the tubes being expanded at their ends into the drums. The upper or steam drum is of large diameter, thus giving ample steam reserve. On its lower side, the plate forming its shell has the tube holes milled out. The mud or lower drum is of smaller diameter, and has on its upper side a corresponding number of holes milled.

These two drums are united by a number of tubes, which are not only vertical, but are all absolutely straight. The tubes are arranged so that in plan they form a continuous W, there being two rows in each arm of the W.

The evaporating tubes (4) are either 2$\frac{3}{16}$ ins. or 2$\frac{3}{8}$ ins. diam. About the centre of the arm of each W, a number of large tubes (5), either 3$\frac{1}{4}$ ins. or 4 ins. diameter, are arranged. In the latter, smaller concentric (6) tubes are supported by means of spiders, not as in the well-known Field tube, in which the external tube is closed at its lower end. In our boiler, the lower end of the outside tube is expanded into the bottom drum; while the lower end of the smaller internal tube projects in the lower drum, somewhat as shown in Fig. 10.

The water is fed into the boiler at the top of, that is, in the steam space above the water level. The incoming water fills an inverted elbow in the boiler, and then flows down distributing trays which run nearly the whole length of the steam drum. The colder water then passes down the downcomer tubes as indicated in the sketch.

THE "NICLAUSSE" WATER-TUBE BOILER—

. . . as supplied by *Messrs. The British Niclausse Boiler Co., Ltd., Clock House, Arundel Street, Strand, London, W.C.2.*

In the Niclausse boilers there is a rectangular battery of 3¼ ins. "field" tubes about 7 ft. long at an angle of 6° to the horizontal, and all the evaporating tubes are fitted at one end only into headers, and a perfectly tight joint is ensured by metal-to-metal conical joints. The other end is closed by a cap, and is carried by a supporting plate. The removal of any tube, either for cleaning or replacement, is very easy and rapid to carry out. The tubes are fixed at one end only, and the tubes are quite free to expand or contract in the flames. Each header is divided into two compartments by a vertical partition plate separating the steam and water currents.

The current of water flowing through the front compartment of the header is distributed to each evaporating tube by an inner tube contained in the former. Openings cut in the header end of the evaporating tube enable the water to enter and the steam to depart in two distinct currents, so that positive circulation in only one, and that in the correct direction, is ensured.

THE "STIRLING" BOILER—

. . . as supplied by *Messrs. The Stirling Boiler Co., Ltd., 58 Victoria Street, Westminster, London, S.W.*

This boiler is made in three standard designs, consisting of 2 steam drums and 1 mud drum, 3 steam drums and 1 mud drum, and 3 steam drums and 2 mud drums.

The standard boiler for all ordinary land purposes is the 5-drum type. The whole boiler rests on steel framework, and the brickwork merely acts as a housing, the mud drums being suspended by the tubes; the safety valves and stop valves are generally connected to the middle or rear steam drum, and feed connections are placed at the end of the rear steam drum. The steam drums in a typical boiler are 4 ft. diameter and mud drums 3 ft., and the drum interiors perfectly clear. The tubes are best weldless mild steel.

With regard to the furnace, by reference to the sectional view of the boiler, it will be seen that a firebrick arch is sprung over the grates immediately in front of the first bank of tubes. The large triangular space above the arch between the boiler front and the tubes is available for combustion chamber. The arch absorbs heat from the fire, becoming an incandescent radiating surface, which heats the air required for combustion, ignites by radiation the gases distilled from the coal, and prevents the boiler from being chilled by an inrush of cold air when the furnace doors are opened.

The circulation is as follows: The feed-water entering the rear top drum through the check valve passes into a feed-distribution box, which is placed the whole length of the drum. By means of this box, the feed is distributed over the whole width of the boiler, and ensures that every tube receives a proportion of the entering feed-water.

The feed-water passes down the rear bank, up the third bank, down the second bank, and up the front bank. Here

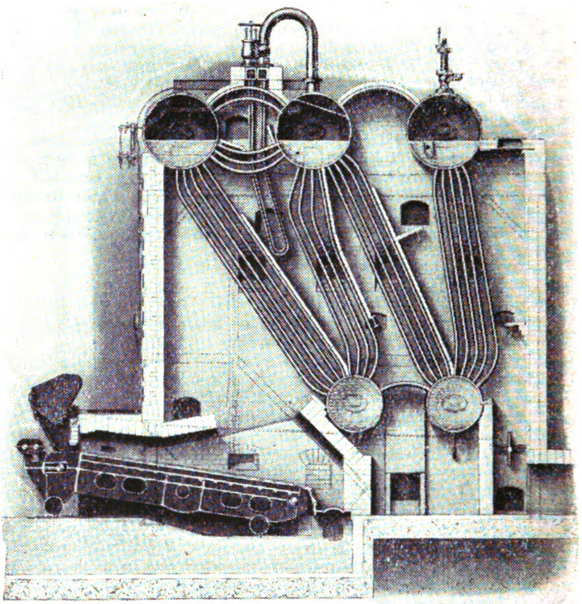

FIG. 11.—STIRLING BOILER.

the steam formed during the passage up the tubes disengages and passes through the upper or steam-circulating tubes into the middle drum, while the solid water passes through the lower or water-circulating tubes into the middle drum. This water again joins the main circulation and passes down the second bank and up the front bank, continuing its former course until it is finally evaporated.

The gases flow in the opposite direction to the feed-water. After combustion takes place in the furnace, the volume of gas passes up the front bank, down the second, up the third, and down the rear to the chimney. Owing to the long distance passed over before the chimney is reached, the most intimate contact is obtained of the hot gas with the boiler surface, and the heat is quickly and thoroughly extracted.

THE "SUCKLING" WATER-TUBE BOILER—

. . . as supplied by *Messrs. Thomas Beeley & Son, Ltd., Hyde Junction Iron Works, Hyde, near Manchester.*

This boiler is made entirely of wrought-steel, and the drums are circular without any flats, and have spherical or dished ends. The tubes are solid drawn, and no stay bolts or braces necessary, the working pressure being 200 lbs. or over.

The boiler is composed of one or more sections, and each section is a complete unit in itself. Each section consists of 2 drums, the upper or steam-drum being horizontal and the lower or water-drum inclined. The course of the gases is that on leaving the furnace they take an upward direction and thence beneath the top or steam-drum, meeting in their course the large frontal connection and the successive banks of vertical tubes, which are "staggered." On arrival at the back-end, the gases descend and return beneath the heat baffles, coming in contact with the lower portions of the vertical tubes and finally pass beneath the lower drum. The heat of the gases, therefore, is reduced to a minimum before entering the flues. The gases at their highest temperature come in contact with the hottest portions of the boiler, and they strike the tubes practically at right angles. (*See* Figs. 12 and 13, p. 650).

THE "THOMPSON" BOILER—

. . . as supplied by *Messrs. John Thompson & Co., Ltd., Wolverhampton.*

This boiler has all the circulating tubes perfectly straight, there being three sets, each coupling a top and bottom header. It is built on mild steel girders resting on mild steel columns independent of the brickwork, and all parts are free to expand. All tubes of solid drawn steel 3¼ ins. diameter.

THE "WOODESON" BOILER—

. . . as supplied by *Messrs. Clarke, Chapman & Co., Ltd., Gateshead-on-Tyne.*

This boiler, as seen from the illustrations (page 651), is made up of one or more sections, each consisting of a steam drum at top, a water drum at bottom, and a number of groups of tubes (the number of groups in each section depending on size of boiler required) expanded into flat discs on the steam and water drums—these flat discs are pressed by hydraulic pressure out of the solid plate.

Each steam drum is connected to its neighbouring steam drum by horizontal connecting or circulating tubes, and the water drums are also connected in a similar manner.

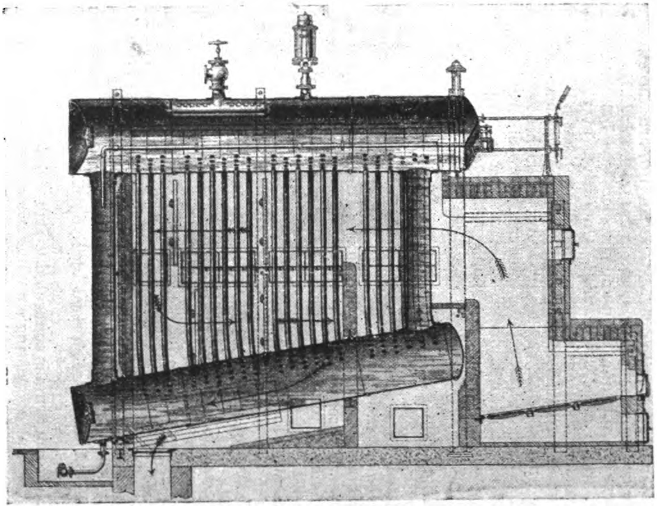

FIG. 12.—SUCKLING BOILER.

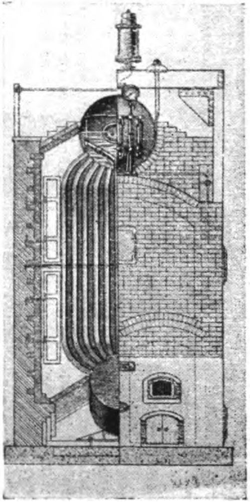

FIG. 13.—SUCKLING BOILER.

The whole boiler is hung suspended on girders, and is entirely free at the bottom to allow for any expansion.

On the top of the steam drums are arranged a series of manholes, one immediately over each group of tubes, and any tube can be withdrawn and replaced through the manhole above the particular group in which the tube is situated.

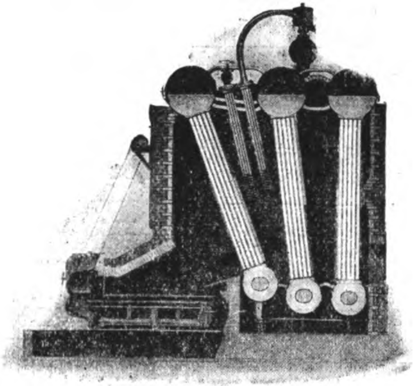

FIG. 14.—WOODESON.

The feed water enters into the steam drum at the point farthest away from the fire, and the water flows down the tubes in the rear section and up the tubes in the front sections.

The advantage of this circulation is obvious, as the water enters the boiler at the coldest part and flows down the rear tubes in contact with the coldest gases, precipitating any mud or dirt into the bottom drum, and all the tubes in the front which are in contact with the greatest heat have practically clean water to deal with, therefore no deposit is found in any of the tubes exposed to great heat.

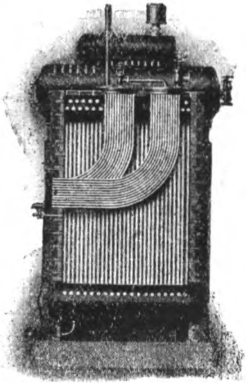

FIG. 15.—WOODESON.

CLARKE-CHAPMAN WATER TUBE BOILERS

(“ *Woodeson's* ” *Patent*)

3-SECTION TYPE. CHAIN GRATE STOKERS

Distin-guishing No.	H.S. sq. ft.	G.A. sq. ft.	No. of Sect.	No. of Groups per Sect.	Length of Tubes.	OVERALL SIZES.			BRICKWORK.		SHIPPING.	
						Height over Steam Drums.	Width of one Boiler.	Front to Back.	Fire.	Ordin-ary.	Tons.	Cubic Measure.
12E	1175	21½	3	2	12′ 6″	20′ 9″	6′ 3″	24′ 0″	3500	12400	24	1500
12F	1225	24	3	2	13′ 0″	21′ 3″	6′ 3″	24′ 6″	3680	12750	25	1550
12G	1260	26	3	2	13′ 6″	21′ 9″	6′ 3″	25′ 0″	3850	13100	26	1600
12M	1350	28	3	2	14′ 6″	22′ 9″	6′ 3″	26′ 0″	4100	13800	27	1710
13E	1760	35	3	3	12′ 6″	20′ 9″	7′ 10″	24′ 6″	4400	14500	33	2070
13F	1820	37	3	3	13′ 0″	21′ 3″	7′ 10″	25′ 0″	4540	14750	34	2120
13G	1890	38	3	3	13′ 6″	21′ 9″	7′ 10″	25′ 3″	4680	15000	35	2170
13M	2025	41	3	3	14′ 6″	22′ 9″	7′ 10″	26′ 0″	5050	15400	36	2260
14E	2350	47	3	4	12′ 6″	20′ 9″	9′ 5″	25′ 0″	5200	15800	38	2360
14F	2450	49	3	4	13′ 0″	21′ 3″	9′ 5″	25′ 6″	5360	15950	39	2400
14G	2520	51	3	4	13′ 6″	21′ 9″	9′ 5″	26′ 0″	5520	16100	40	2440
14M	2700	52	3	4	14′ 6″	22′ 9″	9′ 5″	26′ 3″	5900	16400	41	2570
15E	2970	55	3	5	12′ 6″	20′ 9″	11′ 0″	25′ 0″	6200	17100	45	2760
15F	3050	61	3	5	13′ 0″	21′ 3″	11′ 0″	25′ 6″	6360	17250	46	2830
15G	3185	64	3	5	13′ 6″	21′ 9″	11′ 0″	26′ 0″	6520	17400	47	2900
15M	3450	67	3	5	14′ 6″	22′ 9″	11′ 0″	26′ 6″	6830	17900	48	2950
16E	3500	70	3	6	12′ 6″	20′ 9″	12′ 7″	24′ 6″	7150	18200	51	3100
16F	3650	73	3	6	13′ 0″	21′ 3″	12′ 7″	25′ 0″	7280	19100	51½	3120
16G	3780	76	3	6	13′ 6″	21′ 9″	12′ 7″	25′ 6″	7400	20000	52	3140
16M	4050	79	3	6	14′ 6″	22′ 9″	12′ 7″	25′ 6″	7620	21500	53	3200
17E	4125	83	3	7	12′ 6″	20′ 9″	14′ 2″	25′ 6″	7800	22300	60	3630
17F	4250	85	3	7	13′ 0″	21′ 3″	14′ 2″	26′ 0″	8000	22600	61	3660
17G	4400	88	3	7	13′ 6″	21′ 9″	14′ 2″	26′ 0″	8225	22900	61½	3780
17M	4750	89	3	7	14′ 6″	22′ 9″	14′ 2″	26′ 0″	8720	23400	62	3790
18E	4750	92	3	8	12′ 6″	20′ 9″	15′ 9″	24′ 6″	9000	23800	67	4000
18F	4900	97	3	8	13′ 0″	21′ 3″	15′ 9″	25′ 0″	9550	24150	68	4060
18G	5040	101	3	8	13′ 6″	21′ 9″	15′ 9″	25′ 6″	10000	24500	68½	4150
18M	5400	105	3	8	14′ 6″	22′ 9″	15′ 9″	26′ 0″	10800	25200	69	4200

The fire-grate is arranged across the boiler immediately in front of the front sections, and the gases travel upwards among the tubes of the front sections and over the firebrick baffle arranged between the sections and downwards among the tubes of the rear section and out at the back wall.

A large steam dome is arranged over the steam drums to ensure perfectly dry steam being obtained.

Large doors are arranged in side wall of boiler for access into the bottom drums, and also into combustion spaces for inspection, etc., and small sight holes are also arranged in the walls for inserting a steam jet for clearing the outside of the tubes of soot.

The boiler is fitted with the "Woodeson" patent Superheater, with by-pass to regulate the temperature of the superheat as required.

The boiler is, of course, made in various types.

BOILER MOUNTINGS.

Standard boiler mountings vary a little, depending on the pressure and type of boiler, but the following is typical for a "*Lancashire*" boiler and applies more or less to all boilers. Water-tube boilers especially have slightly different fittings, depending on the type.

SAFETY VALVES.

(*a*) High-steam and low-water valve, which consists of a combined lever weight safety valve and a float, so that on the level of the water in the boiler becoming too low, this valve is released and steam escapes. Also if the steam pressure becomes too high, the steam escapes. Fitted with escape bend.

(*b*) Dead-weight safety valve for high-steam pressure, loaded to 5 lbs. per sq. in. in excess of stipulated working pressure.

FUSIBLE PLUGS.

Although some difference of opinion exists as to the efficacy of these appliances, they are often fitted in the furnace tubes as an additional precaution against low-water dangers.

MAIN STEAM VALVE.

Consisting of 5–7 ins. main junction valve, depending on the size of the boiler, with gun-metal valve and seating-packed gland, hand-wheel, and cast-steel bend.

FEED VALVE.

2–2½ ins. check feed-valve for controlling the inlet of the feed-water, combined with a check or back-pressure valve.

BLOW-OFF COCK.

2–2½ ins. asbestos-packed blow-off cock, with box key for blowing off the water in the boiler, made of brass, and so constructed that the key cannot be removed until the tap has been closed together with special wrought-iron key.

WATER GAUGES.

Two water gauges (¾ in.) for showing the water level in the boiler, and provided with an arrangement so that if a gauge glass bursts the outlet of steam is automatically checked by means of a movable ball arrangement. Gauge taps to be of gun-metal asbestos packed. Also provided with heavy gauge-glass protectors, as a protection to the fireman in case of fracture. Also metal pointer for indicating correct level.

STEAM GAUGE.

One 10 in. steam gauge graduated as required (generally in 5 lbs.) to double the blow-off pressure and the blow-off pressure indicated in red, together with inspectors' gauge connection, and pendant cock and syphon.

ISOLATING VALVE.

In some cases, an isolating valve is fitted, that is, a valve so arranged that if a fracture occurs in a steam pipe, the boiler is shut off by an automatic "isolating" valve working with a loose ball, which rises and closes the outlet.

In addition to the boiler mountings mentioned, additional fittings for steam pipes are reducing valves, surplus valves, and back-pressure valves.

Steam valves are generally of two classes—

1. "Sluice" valves, in which the valve slides on the face of the seating.

2. "Mushroom" valves, in which the valve lifts off its seat.

It has been the general practice to make the diameter of valve the same as the diameter of supply steam pipe to prevent throttling and to make maximum lift equal to ¼ diameter. In practice, a valve seldom opens more than $\frac{1}{12}$–⅛ of its diameter, and the "Hopkinson-Ferranti" valve has only ½ area of diameter of pipe.

Over 400° F., gun-metal tends to become soft, and is then damaged by grit in the steam, so that above this temperature it is better to have special nickel-alloy seats for the valves.

Reducing-valves are very valuable when steam has to be used at different pressures from the same boiler-plant. A great deal of trouble has been caused by certain makes of reducing-valves, and they should always be fitted with a safety-valve as a precaution. Also they should, if possible, always have plenty of room for the steam to act as a "cushion." It is not reasonable to expect a reducing-valve to reduce the pressure to a constant figure a few feet away from the valve. For details, the makers' catalogues must be consulted.

BOILER SETTING AND FLUES

The standard setting for "Lancashire" boilers is as follows—

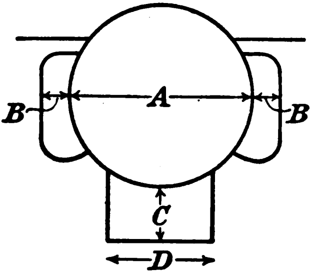

FIG. 16.—BOILER SETTING AND FLUES.

A. Diameter of Boiler.	B. Width of Side Flues at narrowest point.	Depth of Flue under Boiler.	Width of Flue under Boiler.
3′ 6″	12	1′ 9″	1′ 9″
4′ 0″	12	2′ 0″	1′ 9″
4′ 6″	12	2′ 3″	1′ 9″
5′ 0″	12	2′ 6″	2′ 0″
5′ 6″	12	2′ 9″	2′ 0″
6′ 0″	12	3′ 0″	2′ 3″
6′ 6″	12	3′ 3″	2′ 3″
7′ 0″	12	4′ 0″	2′ 6″
7′ 6″	12	4′ 3″	2′ 6″
8′ 0″	12	4′ 6″	2′ 6″
8′ 6″	12	4′ 9″	2′ 9″

The boiler should be set with 1½ ins. drop to the front-end to allow of all the water being drained off by the blow-off cock.

Flue-gases, on leaving the fires, should pass under the bottom of the boiler, splitting at the front-end and flowing along each side-flue to the main flue. By this means, bottom and sides of boiler kept at a more equable temperature than if gases sent reverse way, namely, side-flues first.

The top of side-flues not to be above the furnace crowns, and the flues must be arranged to have good drainage, so that no water can lodge. Neglect of this is apt to cause serious damage, due to corrosion to boilers and economizers.

Great care must be taken with the foundations to avoid subsidence, and in most cases the best practice is to erect the boiler and foundations on a "raft" of concrete at least 12 ins. thick.

There is always trouble with leaky brickwork, due to expansion and contraction of the boiler, and very serious losses in efficiency are caused in this way. The flues of the boiler, and in most cases the economizer, should be lined with 4½ ins. fire-brick, and fireclay should be used instead of lime mortar.

To minimize this leakage, cement over the walls and then paint. Also the brickwork can be "painted" with either a mixture of mortar and cement, or boiler tar and cement heated up when ready for use.

A considerable number of improved types of seating blocks, flue covers, downtake covers are now on the market.

The object of these is to make a much tighter joint than the old-fashioned seating block, together with much less bearing surface. For example, the old seating block has about 4 ins. of bearing surface in contact with the boiler-plates, whereas the modern types have only about 1 in. This improves the heating area of the boiler, tends to easier examination, and reduces the danger of corrosion.

The seating arrangement for water-tube boilers and the various other types of boiler on the market are of a specialized nature to suit the particular boiler, and full particulars are supplied by the boiler makers. The area of the main flue of a boiler-plant must be sufficient to reduce the velocity of the flue gases to not more than about 1,200 ft. per minute calculated on, say, 1 lb. of coal equals 21 lbs. of flue-gas. From another point of view, the area should be at least 20–40 per cent. greater than the chimney area. The designing of flues is more a matter of experience than formulae, and most flues are too small for efficiency. A rough formula is—

$$A = \frac{NT}{144{,}000}$$

A = Area in sq. ft.
N = Total coal burnt per hour in lbs. on all boilers.
T = Absolute temp. of gases leaving the boiler.

In general, main flues should be rectangular, with a width of, say, 0·65–0·85 per cent. of the height, and the top can be flat, arched, or crowned. A very convenient top is flat cast-iron plates, which can easily be taken up and replaced.

MECHANICAL FIRING

Mechanical stokers are of two general types, namely, "coking" and "sprinkling."

In a "coking" stoker, the volatile hydrocarbons of the coal are burnt at a different portion of the grate from the fixed carbon; that is to say, a "coking" stoker first "cokes" the coal, burning off the volatile matter, and then subsequently, on the further portion of the grate from the boiler front, burns the coke produced.

In a "sprinkling" stoker, the coal is thrown in small quantities all over the grate by means of a throwing shovel, and the complete length of the grate is utilized for the combustion of the volatile matter and the fixed carbon of the coal at the same time.

Mechanical firing is of very ancient origin, and the first embryonic mechanical stoker was invented by James Watt in 1785, whilst the first practical stoker was that of William Brunton in 1819. The chain grate stoker was invented by John Juckes in 1841, and consisted of longitudinal firebars connected by links forming an endless chain. The coal was deposited on this travelling chain from a hopper in front of the boiler, and

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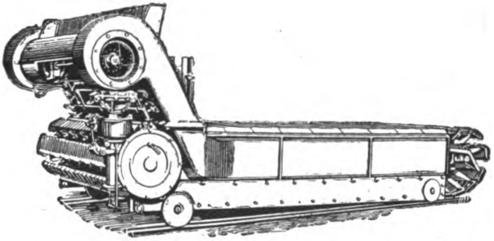

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the fuel gradually burnt as the grate travelled forward discharging ash and clinker over the back. The modern chain grate stoker (p. 646) for water-tube boilers is simply a development of this stoker.

The first sprinkling stoker was invented by John Stanley in 1822, and consisted essentially of two horizontal fans revolving on vertical spindles, which scattered or sprinkled the coal in small quantities all over the grate so as to imitate hand-firing, and was developed by Dillwyn Smith and Henderson, 1870–1875.

The Author proposes to describe only the more generally used types of mechanical stoker in Great Britain to-day, as lack of space prevents a description of foreign stokers and others of historical interest. It is convenient also to divide mechanical stokers into two general types based on the boiler in use, which are as follows—

A. *Narrow Grate or Cylindrcial Internally Fixed Boilers*, such as "Lancashire," "Cornish," "Marine," and all similar types of boilers, in which the grates do not exceed 3 ft. 6 ins. in width.

B. *Wide Grate Boilers*, being water-tube boilers, where the width of the grate may be anything up to 12 or 15 ft., depending on the size of the boiler, externally fixed.

Taking narrow grate boilers first, the makes of "coking" stoker in general use to-day in Great Britain are the "Bennis" (coking), "Hodgkinson," "Meldrum (Koker)," and "Vicars."

The general principle of all these stokers is the same. The coal is fed into a hopper in front of the boilers, and falls down in front of a pusher, which is actuated by driving gear. The "pusher" feeds the coal on to the front of the grate, piling it up in a thick mass in which the volatile matter is burnt off. The grates are fitted with moving bars, with a to-and-fro motion actuated by driving gear, and the burning fuel is gradually carried the whole length of the grate until it falls over the end as ash. The bars are provided with a steam jet apparatus to keep the bars cool, to prevent burning out or warping, and in some cases to provide a forced draught. These stokers work equally well with chimney draught or mechanical-induced draught. There is, of course, many differences in detail between the various types.

A special type of coking stoker is the underfeed type, namely, the "Underfeed" stoker and the "Erith" stoker. In this type of stoker, the coal is forced up from underneath the grate by means of a ram or worm conveyor. The principle is, therefore, that of "underfeed" instead of the usual "over-feed," in which the coal is supplied to the top of the grates.

The grate of the underfeed type of stoker is really a species of trough in which the volatile matter is burnt on the apex of a pile of burning fuel, and the coke burnt as it travels upwards and sideways through the whole length of the trough. These stokers will not work with chimney draught, but only with mechanical forced draught, which is provided as part of the stoker equipment.

The makers of "sprinkling" stoker in general use are the

"Bennis (sprinkler)," Meldrums (sprinkler)," "Proctor," and "Triumph." In these stokers the coal from the hopper falls down in front of a "shovel," which jerks or scatters the coal all over the grate, the shovel being worked by means of springs in connection with the driving mechanism. The grates are fitted with moving bars, with a to-and-fro motion actuated by the driving gear, and the ash gradually falls over the end of the grate. The bars have a steam jet arrangement to keep them cool to prevent warping, and this type of stoker works equally well with chimney draught or induced mechanical draught.

As regards wide grate boilers (water-tube boilers), in the first place the ordinary stokers for narrow-flued boilers can be used side by side, and this is sometimes done. The general custom is, however, to have special stokers for the purpose, and they are all of the "coking" type. One general design is the chain grate stoker, as "Babcock & Willcox," "Bennis," and other makes. An adaptation of this is that of travelling bars such as the "Underfeed" travelling grate stoker. A special type of stoker is the "Niclausse," whilst the underfeed types are represented by the "Erith Riley" stoker and the "Underfeed." The principle is exactly the same as the "coking" stokers for narrow-flued boilers.

A short description of each of these stokers is as follows, and further information can be obtained direct from the makers—

A. Narrow Grate Boilers

1. COKING STOKERS (OVERFEED)

THE "BENNIS" COKING STOKER—

. . . as supplied by *Messrs. Edward Bennis & Co., Ltd., Little Hulton, Bolton.*

In the latest type of this stoker, underneath the coal hopper are cut-offs which can entirely isolate the hopper of coal from the fires when it is desired to stop the machine. The stoker is fitted with duplicate feed-boxes with reciprocating feeder rams, which feed the coal into the furnaces alternately. The mechanism for working the pusher bars is a special cam of doubled chilled steel controlled by hand nuts, so that the feed of the coal can be adjusted from almost zero to an ample overload. The furnace, instead of the usual deadplate in front of the bars, has a series of small bars.

The main furnace bars consist of reciprocating bars supplied with air under pressure by means of steam jets. As the coked fuel leaves the special dead plate, it has to fall several inches on to the grate proper, and this breaks up the mass of partially coked fuel so as to render the combustion more efficient. The ash falls over the back in the usual manner.

THE "HODGKINSON" STOKER—

. . . as supplied by *Messrs. James Hodgkinson (Salford), Ltd., Ford Lane Works, Pendleton, Manchester.*

Coal is fed either by hand or automatically into the hoppers, which have for their base a ram or pusher, over the firedoor, sliding forwards and backwards. On the outward movement, an amount of coal which is always under control, falls into the ram boxes, being pushed slowly on the return of the ram on to the bottom coking or dead plate, being spread in an approved manner over the whole width of the furnace by means of the top coking or distributing plate. Here it becomes a mass of coke, the free gases and volatile matter, being liberated and passing over the whole length of a well-covered and incandescent firegrate, becoming entirely consumed. The action of the firebars—operated by a special form of hardened cam—ensures the continuous but slow travel of the fire to the back end and the air spaces being always open. The firedoors remaining closed and the feed of fuel entirely covering the grate surface without outside help.

"MELDRUM" COKING STOKER—

. . . as supplied by *Messrs. Meldrums, Ltd., Timperley, near Manchester.*

In this stoker, the front is of hollow construction and usually air-cooled by means of a steam jet, so that the boiler house is kept cool. The ram is of a patented rector design and, as usual, the travel can be adjusted. It is completely enclosed to avoid leakage of dust. The firebars are of a patented design, and consist of a light framework into which are fitted small-filling bars. These are interlocked, so that they cannot be accidentally displaced. Should the small bars burn, which is a very rare occurrence, owing to the large cooling surface, the cost of renewal is trifling.

The bars are spaced with narrow air spaces, so that the loss by small coal falling through is very slight. The bar ends which push the clinker into the ash pit are renewable, and the faces upon which the cams work are also fitted with renewable nibs of chilled cast-iron. The cams working the bars of cast iron with chilled faces and the gear boxes are enclosed. This stoker will work with ordinary mechanical forced or induced draught, and is often fitted with "Meldrums" steam-jet forced draught.

THE "VICARS" STOKER—

. . . as supplied by *Messrs. T. & T. Vicars, Earlestown, Newton-le-Willows, Lancashire.*

The stoker is supported entirely from the ground, and is not hung on the boiler front. The fuel is fed from the hoppers in the usual way, and is gradually pushed by rams, working alternatively, over the coking plates and then on to the dead plates, where it cokes before passing on to the moving firebars. All the bars travel inwards together towards the bridge; on returning, however, each alternate bar is locked in position, while the remainder are being withdrawn. The coal is therefore carried in the furnace by the simultaneous forward action of all the bars, but is not carried back by the bars because the fresh fuel coming up behind prevents this. The burning coal

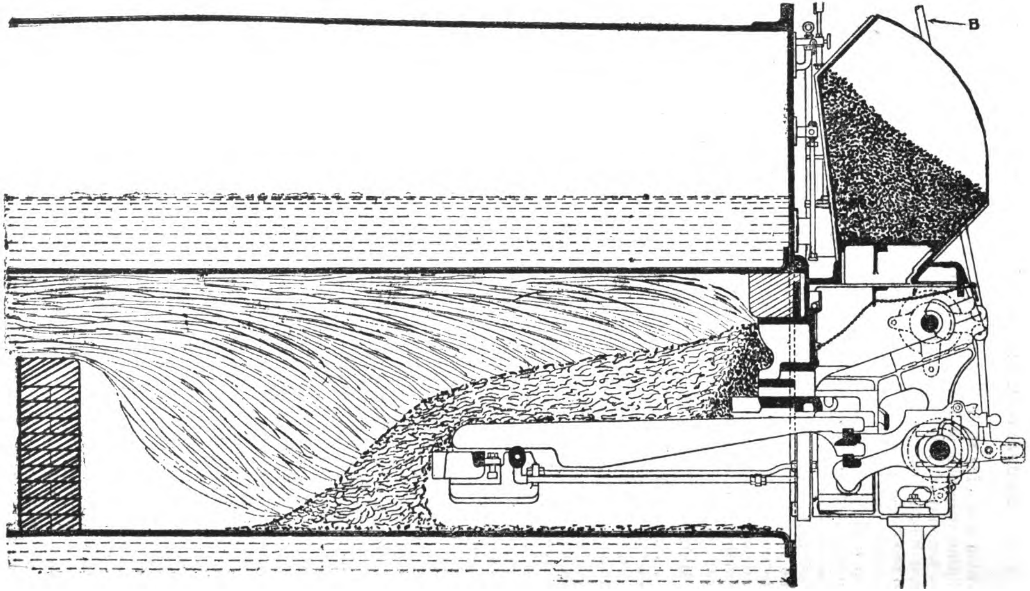


FIG. 17.

is thus gradually moved forward as it burns, until it falls as ash over the end of the bars. The amount of feed of the coal and of the travel of the bars can be adjusted by suitable control levers. This stoker is specially suitable for working with induced draught.

Grates generally are 6′ 6″ × 3′ 6″, and air space between the bars ⅛″–⅜″. Power to drive very small, and makers state that 1 B.h.-p. will drive several machines. Bars are kept cool with a small amount of steam in the form of jets under the back of the bars.

2. COKING STOKERS (UNDERFEED)

The "Underfeed" Stoker—

. . . for "Lancashire" boilers, as supplied by *Messrs. The Underfeed Stoker Co., Ltd., Coventry House, South Place, Finsbury, London, E.C.*

The principle of the "Underfeed" Stoker is as follows:

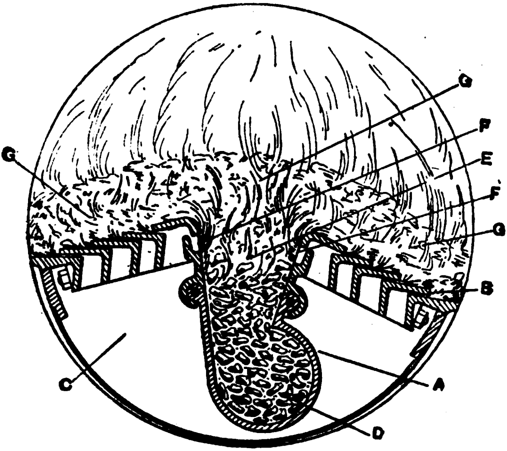


Fig. 18.—Cross-section of the Stoker in the Flue of a Lancashire Boiler.

Fig. 18 (above) shows the cross-section of one of the flues of a Lancashire boiler, containing the stoker illustrated on the following pages. *A* is the retort or fuel magazine, in the lower or circular part of which revolves a taper feeding worm which conveys the coal. *B* shows the terraced grates, *C* the wind box, and *D* the green coal, which, being gradually pushed up to the burning point, has commenced to ignite at the point *E*, where it meets the incoming air from the tuyeres *F* and F^1. Here the coal is coked (i.e. deprived of its volatile hydrocarbon gases). These being mixed with air introduced at this point through the air inlets of the fuel magazine, and escape being possible only by rising through the glowing coke above, they are heated to such a high temperature that they are completely consumed without smoke, while the combustion of the fixed carbon of the coal is completed by the air introduced through the apertures in the sides of the terraced grates *B*. The effect of this method of underfeeding is to keep a perpetually clear bright surface of incandescent fuel, which always produces its maximum steaming effect upon the boiler without the inevitable fluctuations of furnace temperature consequent upon hand-firing.

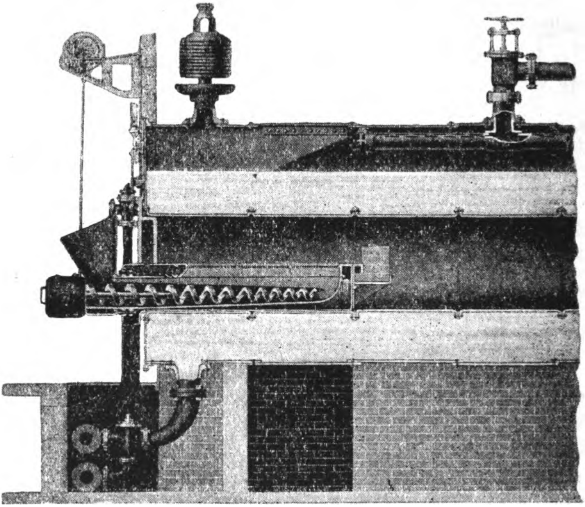


FIG. 19.—LONGITUDINAL SECTION OF THE GEAR-DRIVEN UNDERFEED STOKER IN FLUE OF A LANCASHIRE BOILER.

3. SPRINKLING STOKERS

THE "BENNIS" STOKER—

. . . as supplied by *Messrs. Edward Bennis & Co., Ltd., Little Hulton, Bolton.*

In this stoker, under each hopper, is a cast-iron feeding-box, in the interior of which is a pusher plate with an adjustable reciprocating motion. The fuel from the hopper falls in front of the pusher plate and is pushed over a ledge formed by the bottom of the feeding-box. The amount of fuel pushed over is regulated in a simple manner by means of an adjustable cam and the turning of a hand-nut whilst the machine is in motion. The variation obtained is very great, being anything from practically zero to, say, 1 ton per hour.

The fuel then falls on a flat plate called the shovel box, and is scattered into the fire by means of an angular shovel. This shovel is worked by a spring, which is enclosed in a cylinder, and the shovel is so arranged that the coal is only thrown on a quarter of the fire at once. The stoker is fitted with the usual moving-bar arrangement, fitted with steam jets to supply forced draught. These moving bars consist essentially of tubular troughs containing short interlocking grate bars or grids about 2 ft. long, rounded at the top to prevent clinker sticking. The complete bars have a to-and-fro motion so arranged that each bar on its outward journey moves between two other bars which are at the moment stationary, and the coal travels steadily along the length of the furnace until it falls over the back as ash.

"MELDRUM" SPRINKLING STOKER—

. . . as supplied by *Messrs. Meldrums, Ltd., Timperley, near Manchester.*

This stoker can be supplied with ordinary moving bars, or with a fired grate similar to hand-firing. This "sprinkler" stoker is on the same general lines as the "coking" stoker already described, the front being hollow and air-cooled as usual, and the cams and gear-box being similar. The coal is thrown by means of a "sprinkler" of the usual design, which can be adjusted whilst working; and the general principle will be seen from the illustrations, whilst the firebars are similar in design.

THE "PROCTOR" STOKER—

. . . as supplied by *Messrs. James Proctor, Ltd., Hammerton Street Ironworks, Burnley, Lancs.*

This stoker is so arranged that, by the action of a spring shovel and tappet, it distributes a small quantity of coal to different parts of the fire, viz., to the back, middle, and front, which is effected by the tappet having three different throws. It can also be adjusted to meet any desired length of bar, keeping a level fire the full length of the grate. The distribution of the fuel is effected by means of radial shovels working inside boxes with specially curved bases to give the proper direction to the fuel, and there are no cranked joints in connection with the operating levers. The moving bars are operated by an eccentric and rack-wheel motion, so that the speed of the travel of the bars can be adjusted to suit individual conditions. The quantity of coal to be burnt can be regulated by simply adjusting the regulating screw of the ram, which supplies coal to the shovel. The amount can be from, say, 2 cwt. of coal per furnace per hour up to almost any quantity, according to the draught available and the quality of the fuel. This stoker works equally well with mechanical or natural draught.

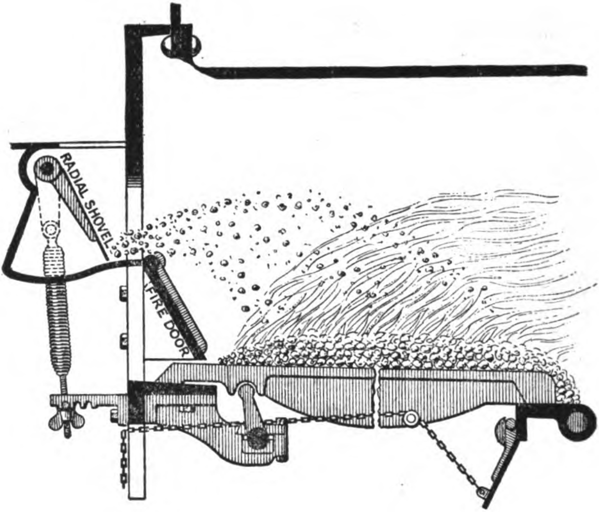


FIG. 20.—PROCTOR SHOVEL STOKER.

THE "TRIUMPH" STOKER—

. . . as supplied by *Messrs. The Triumph Stoker, Ltd.*, 85 *Kirkstall Road, Leeds.*

The coal is thrown or sprinkled over the fire by the automatic shovel at intervals of a few seconds, and no two successive charges fall in the same place.

The coal falls from the hopper into a box containing an adjustable pusher slide which has a slow reciprocating motion. At regular intervals the slide allows a small quantity of coal to fall in front of the "shovel," which is worked by a trigger and spring air-cushioned. The "shovel" throws the coal over the fire, and the speed can be adjusted as required. The coal as it burns, travels along the bars and over the end as ash in the usual manner by the "to and fro" motion of the bars, worked with a cam arrangement under the front of the fires.

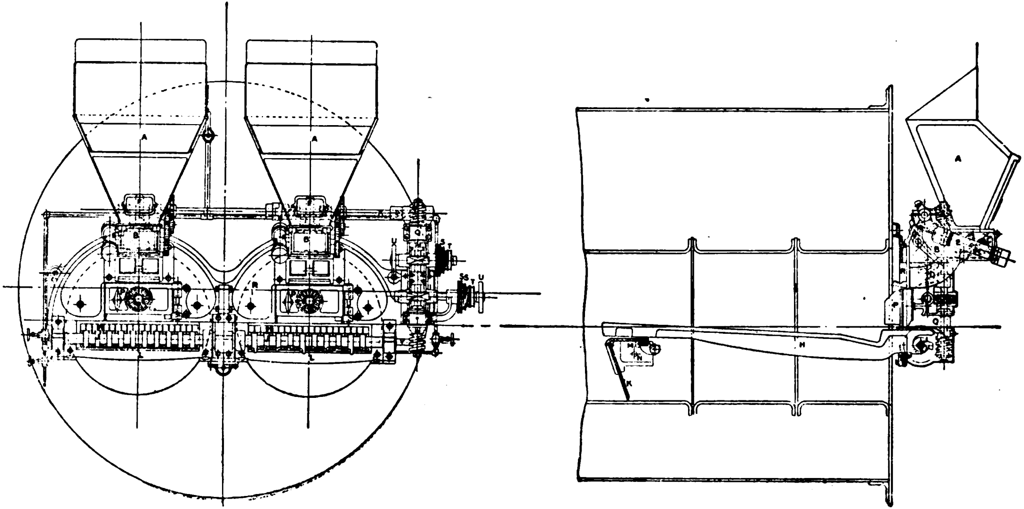


FIG. 21.—TRIUMPH STOKER.

B. WIDE GRATE BOILERS (WATER-TUBE BOILERS)

THE "BABCOCK AND WILCOX" CHAIN GRATE STOKER—

. . . as supplied by *Messrs. Babcock & Wilcox, Ltd., Oriel House, Farringdon Street, London, E.C.*

The grate consists of an endless chain of short interlocking cast-iron grate bars, linked together, running on rollers, and driven by a revolving drum at the front end of the stoker. The necessary power to drive the revolving drum is transmitted through a link chain and sprocket wheels from a shaft arranged either overhead or underground. The starting or stopping of the stoker, or the variation of speed, is effected through the Babcock & Wilcox patent gear box, which can be driven at varying speeds. This box forms an oil bath in which the gear runs, and also contains an efficient clutch for putting the driving gear out of action should any unusual strain arise from an obstruction on the grate. *See* Fig 9, p. 645.

Any small repair, such as replacing a link, can be carried out with the stoker in place under the boiler; but in case of need, the whole stoker can be brought out clear of the boiler, the frames being provided with wheels running on rails placed at the sides of the ashpit.

The slack of the chain can be taken up from the front end of the stoker, thus avoiding the necessity for obtaining access to the back end for tightening the grate.

The grate bars or links are of a special interlocking pattern, which allow the smallest coals to be used and, while providing for an ample supply of air for the combustion of the fuel, prevent the influx of cold air at the back of the grate. The coal is fed over the whole width of the grate, and the depth of the fire is regulated by the adjustment of the vertically-lifting firedoor.

The feed of the coal is slow, and the gas evolved from the fresh fuel as it drops on to the grate readily perfects its combustion by passing over the incandescent fuel further on.

The depth of the fire being regulated by a vertically-lifting firedoor, the stoker can be adjusted in a moment to suit the class of fuel to be consumed and varying demands for steam, both by the depth of the fire and the speed of the grate. Where the special requirements of the work make it desirable, the stoker can be fitted with an arrangement of vanes for cutting off the air supply to a portion of the grate during periods of light load, thus enabling the general efficiency to be kept up under all conditions of work.

THE "ERITH ROE" UNDERFEED STOKER—

. . . as supplied by *Messrs. Erith's Engineering Co., Ltd., 83 Kingsway, London, W.C.2.*

This stoker is designed for water-tube boilers, and each stoker is composed of any desired number of standardized self-cleaning underfeed retort units, each with its individual safety device. Each unit introduces fresh coal into its individual retort by a slow-moving ram, so that the coal overflows from the retort and over the fuel supporting surfaces or tuyeres, though the air for combustion is correctly distributed. The lower layer of the thick fire-bed is the low temperature coking zone, in which the volatile hydro-carbons are distilled before they pass through the upper layer of burning fuel, wherein combustion is smokelessly completed at high temperature, the incandescent surface of the entire fuel bed radiates heat direct to the water-tubes which form the roof boundary of the furnace.

The "Riley" stoker is a fully automatic underfeed stoker. Its continuous automatic working accounts for the extreme simplicity of the design. It consists of any number of uniform, standardized, inclined retort units. Coal is fed by a single ram to each retort, all rams being actuated by a slow-moving crankshaft. Crossheads on the rams reciprocate the entire fuel-bearing surfaces, with an adjustable movement to suit the fuel. This continuous movement keeps the thick bed of fuel open and porous, and discharges the incombustible ash in a steady stream.

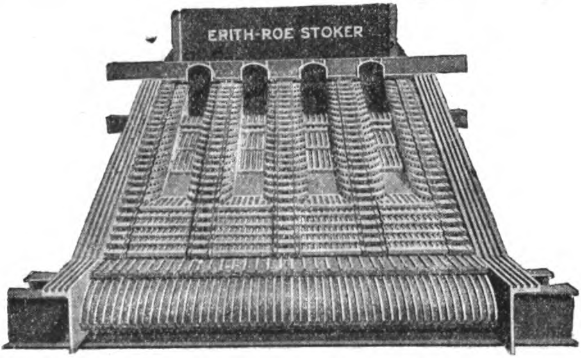


FIG. 22.—THE ERITH-ROE STOKER.

Fig. 22 shows a reciprocating unit, viz., the side plate of a retort, to which the tuyeres admitting air are bolted, and the shoe on which the side plate slides. On this sliding shoe the rocker bar is hinged, its combined horizontal and vertical movement agitates, crushes, and discharges the ash, preventing any accumulation of large clinkers. A pair of these units, together with the fixed bottom plate and base plate, constitutes the automatic retort unit.

The novel feature is that it is provided with independent control for each of the three stages of combustion, that is the coking, main combustion, and burning off the ash, portions.

As usual, no fire-brick arches are required, so that the radiant heat is transferred direct to the boiler tubes and the wear and

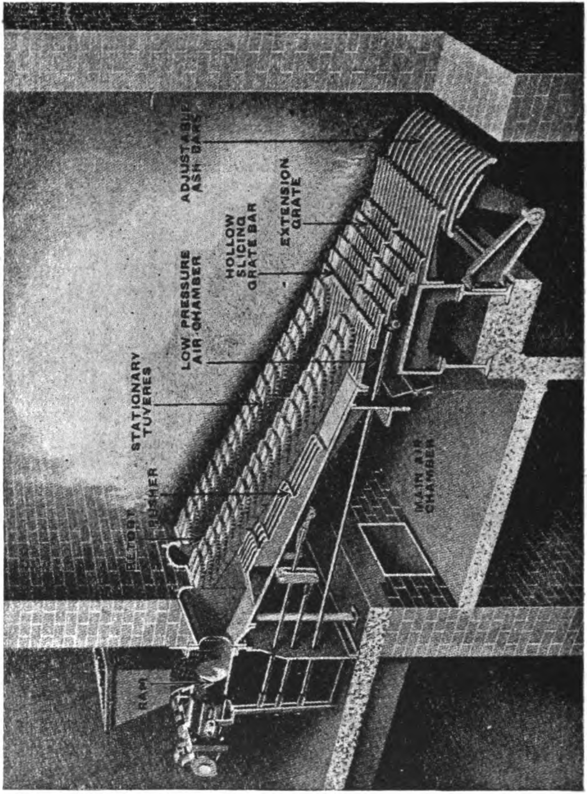


FIG. 23.—ERITH-ROE STOKER.

tear is much reduced. The stoker is self-cleaning, and sufficient units can be assembled to give a single furnace capable of burning 250 tons of coal per day.

The "Underfeed" Self-Contained Impelled Draught Travelling Grate Stoker, Class A—

. . . is the latest type of mechanical stoker supplied by *Messrs. The Underfeed Stoker Co., Ltd., Coventry House, South Place, London, E.C.2.*

This stoker is an adaptation of the well-known "Underfeed" travelling grate stoker arranged so as to bring together in one self-contained unit, the fans, shafting, and driving mechanism. Each machine is entirely independent as regards drive and mechanical draught, and the air, instead of being supplied through a long duct common to all the stokers of a boiler-plant, is supplied by a fan and motor embodied in the mechanism of the stoker itself. The air supply to the grate is also divided into compartments, so that at the end of the travel the amount of air is reduced to correspond to the thin fires, and so maintain a high figure for CO_2. The power required for driving the fan and the stoker is extraordinarily small, less than 0·5 per cent. of the steam production of the plant.

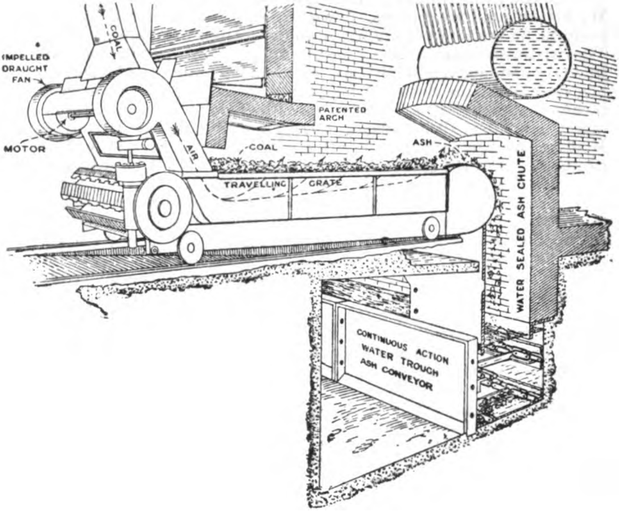


Fig. 24.—Underfeed Impelled Draught Travelling Grate.

GENERAL REMARKS ON MECHANICAL STOKING

There is a very great difference of opinion amongst engineers as regards the advantages and disadvantages of mechanical stoking. In considering this point, there is a sharp distinction between narrow grate boilers and wide grate boilers. In the case of wide grate boilers, mechanical stoking is essential, because the grates are too large to be fired by hand on account of the great size of the boilers. Any disadvantages of mechanical stoking, therefore, for this type of boiler have to be endured. With narrow-flued boilers, however, the case is different, and hand-firing is more commonly employed than mechanical firing. The writer estimates that roughly 25 per cent. of the narrow-flued boilers of Great Britain are mechanically fired, and he has considered in detail the advantages and disadvantages of mechanical firing as applied to narrow-flued boilers in a recent paper read on 19th March, 1920, before the Institution of Mechanical Engineers in London ("Exact Data on the Performance of Mechanical Stokers as Applied to 'Lancashire' or other Narrow-flued Boilers"). This paper gave the complete figures for the performance of eighty typical mechanically-fired "Lancashire" boiler plants in Great Britain.

The net working efficiency of the eighty plants was 59 per cent. as compared with about 62 per cent. on 350 hand-fired plants; and the general idea that as boiler plants are generally worked to-day, mechanical firing is giving better results than hand-firing, is a complete fallacy. With mechanical firing, the advantages were slightly better CO_2, somewhat less smoke, and much improved conditions in the firehole, together with easy mechanical conveying of the coal. As regards saving in labour, this only applies to large plants of eight boilers or over. For 6–8 boilers, the saving is slight, and for four boilers or less there is no saving. It is a much disputed point whether mechanical stokers will in general burn inferior coal better, give more steam, and respond better to fluctuations in steam demand.

Obvious disadvantages of mechanical firing are lack of "flexibility" in the coal burned (that is, certain stokers will only burn certain kinds of coal, and the market available is therefore limited), capital outlay, interest and depreciation, the cost of up-keep, and the amount of steam used by the steam jets. The Author estimates the average cost of upkeep is at least £40 per boiler per annum above hand-firing, and it may be much more. The steam used by the steam jets is a very serious item. The Author has shown that the average figure is about 6·7 per cent. of the total steam production of the plant, and varies anything from 0·5–21 per cent. The figure usually mentioned in this connection is 1–2 per cent. of the production.

The following table will be of interest showing the results of an examination of seventy-six plants, representing three types of sprinkling stoker and five types of coking stoker—

STEAM JETS AVERAGE RESULTS FOR MECHANICAL FIRING

Averages 6·7 per cent. of the production

Type of Apparatus.	No. of Plants.	No. of Boilers	Percentage of Production.	Total Coal Bill.	Total Coal Bill used by Jets.
			per cent.	tons per ann.	tons per ann.
Type A : Sprinkling	25	25	5·00	140,345	7,017·2
„ B : „	16	45	5·25	95,550	5,016·4
„ C : „	7	23	5·00	30,070	1,503·5
„ A : Coking ..	4	12	2·30	21,050	484·1
„ B : „ ..	1	3	13·80	5,750	793·5
„ C : „ ..	13	66	8·00	221,950	17,756·0
„ D : „ ..	1	3	7·20	4,900	352·8
„ E : „ ..	9	63	7·50	185,250	13,893·7
	76	288		704,865	46,817·2

It is a very difficult matter to decide as to whether mechanical firing or hand-firing is the best for a given narrow-flued boiler plant, and many factors have to be studied. Also if mechanical firing is advisable, the choice of the particular stoker is still very difficult. On wide grate (water-tube) boilers, the chief disadvantages are the cost of upkeep and the restriction in the fuel available for good results. Speaking very generally, however, mechanical stokers on water-tube boilers work much better than on narrow-flued boilers.

COAL- AND ASH-CONVEYING PLANT

For a large boiler installation, mechanical coal and ash handling is a great improvement, not only from the point of view of saving in labour, but also because of the greatly improved conditions of the whole plant. With mechanical stoking, generally speaking, coal-conveying plant should be installed on even the smallest plant.

There are very many different types of conveyors on the market, and it is a matter of some difficulty to decide which type is the best for any individual plant.

Conveyors can be classified as follows—

1. Spiral conveyors, consisting of a steel spiral working in a shallow steel trough along which the coal or ash travels.
2. Steel-plate push conveyors, in which a series of projecting plates fastened to a travelling rope pull the coal or ash along a trough.
3. Belt conveyors, in which the coal is discharged on to a moving belt of rubber or metal.

4. "Bucket" conveyors, the essential features of which are a series of small buckets attached to a moving chain, which are automatically filled and discharged as they travel along.

5. Pneumatic elevators, in which the coal or ash is sucked through a pipe by means of a vacuum in a container maintained by a vacuum pump.

6. "Telpher" arrangements, in which a large travelling receptacle is filled and emptied as required, working by means of a travelling wire rope.

Most of the mechanical stoker makers (pp. 656–669) supply varieties of coal- and ash-elevating plant along with the stoker installation, and the following list includes most of the important firms who specialize in conveying plant—

Messrs. The Conveyor and Elevator Co., Ltd., Accrington, Lancs.; Fraser Chalmers & Co., Ltd., Erith, Kent; Goodhall, Clayton & Co., Ltd., Hunslet, Leeds; W. J. Jenkins & Co., Ltd., Beehive Works, Retford; Mechans, Ltd., Scotstoun Ironworks, Glasgow; Henry Simon & Co., Ltd., Mount Street, Manchester; Spencer & Co., Ltd., Melksham, Wilts.

BOILER PLANT ACCESSORIES (FUEL)

1. CHIMNEYS

THE THEORY OF CHIMNEY DRAUGHT.

The draught in a chimney is caused by the difference in weight of the hot air in the interior of the chimney, or smoke stack, compared with the cold air outside. Theoretically, in the first place, the higher the chimney, the stronger the draught. Secondly, the draught depends on the temperature in the chimney. Theoretically, the area inside the chimney does not matter, and a chimney of a given height and temperature will give the same draught, no matter what the area may be.

In practice, several considerations enter into the problem. As the chimney is standing in cold air, the hot gases are "cooled" inside the chimney; and, roughly speaking, under normal conditions, the gases will be cooled about 2°F. for each 3 ft. of height. In calculating the draught, therefore, the mean temperature of the whole of the chimney must be taken, because of this cooling action. This depends on the height of the chimney apart from the condition of the air outside, that is the temperature and the state of the wind. Also there is the action of the friction of the gases on the sides of the chimney. In practice, the layer of flue gas next to the bricks hardly moves at all because of this. It is customary to assume that a layer of gas 2 ins. deep next to the bricks does not move.

A chimney is a wasteful and unscientific method of producing a draught, and the amount of draught simply depends on the waste of heat. Very roughly, an average chimney under average conditions works best with a base temperature of 550–600° F. Professor Rankine says a chimney gives the best results from the point of view of draught production, when the ratio of the absolute temperature of the atmosphere to the absolute temperature of the gases is as 12 is to 25; that is at 50° F. atmosphere, the chimney base temperature would be 602° F.

CHIMNEY FORMULAE.

Many empirical formulae are used by engineers in connection with chimneys, especially to determine the height and area for any given conditions, and also the draught obtainable. It is obvious that any such formulae must be empirical, and can only be regarded as a rough guide. The question is complicated by the gradual cooling of the flue gas in the chimney by the cold air outside, and the friction of the gases inside against the sides of the chimney. Also the surroundings, such as hills, exposure to the wind, or high buildings complicate the position. Further, the design and area of the flues, the great area, air space between the firebars, the thickness of the fires, and the quality of the coal are all important factors.

FORMULAE FOR THE HEIGHT OF A CHIMNEY.

Most of these formulae are based on Peclet's original formulae:

$$h = \frac{U^2}{2g}\left(1 + G + \frac{ft.}{m}\right)$$

h = a height of hot gases which, added to the column of hot gas in the chimney, would produce the same pressure at the boiler fires as a column of outside air, of the same area and height of the chimney.
U = velocity of gases in the chimney.
G = a constant representing the resistance due to the passage of air through the fires.
l = length of flues and chimney.
m = mean hydraulic depth (area of a cross-section divided by the perimeter).
ρ = another constant depending on the roughness of the surfaces over which the gases pass.

Rankine's formula, based on the above formula of Peclets by giving certain values to the constants—

$$h = \frac{\frac{t}{t^2}\left(0{\cdot}0807\right) H - H}{\frac{t}{t_1}\left(0{\cdot}084\right)}$$

$$= \left(0{\cdot}96\frac{t_1}{t^2} - 1\right) H$$

H = height of chimney in feet
t = absolute temperature 493° (that is, melting ice, 32° F.).
t_1 = absolute temperature, gases in chimney.
t_2 = absolute temperature outside air.

Thurston's formula is based on the amount of coal to be burnt per square foot of grate area per hour:

$$H = \left(\frac{F + 1}{2}\right)^2$$

where H = height of chimney in feet.
F = fuel to be burnt in lbs. per sq. foot grate area.

Thus, for example, to build a chimney for four "Lancashire" boilers, each of 36·00 sq. ft. grate area, coal burnt at rate of 25 lbs. per sq. ft. grate area per hour, and temperature in chimney base of 480° F. :

$$H = \left(\frac{F + 1}{2}\right)^2$$

$$= \left(\frac{25 + 1}{2}\right)^2$$

$$= 13^2$$

$$= 169 \text{ ft., say, } 170 \text{ ft.}$$

Another rough rule is to make the height of the chimney three times the length of the boiler plus twice the distance of furthest boiler to chimney.

Wood (*American Society of Mechanical Engineers' Transactions*), based on similar formulae, gives the following table for three different rates of combustion per square feet grate area—

		Height of chimney in ft.		
Standard Temperation of outside air 59° F.	Temperation of Chimney Gases.	Coal burnt per sq. ft. grate area per hr.		
		24 lbs.	20 lbs.	16 lbs.
	239° F.(700 ab.)	250·9	157·6	67·8
	339° F.(800 ab.)	172·4	115·8	55·7
	539° F.(1000 ab.)	149·1	100·0	48·7
	639° F.(1100 ab.)	148·8	98·9	48·2
	739° F.(1200 ab.)	152·0	100·9	49·1
	939° F.(1400 ab.)	159·9	105·7	51·2
	1139° F.(1600 ab.)	168·8	111·0	53·5
	1539° F.(2000 ab.)	206·5	132·2	63·0

Kent's formula, based on actual observations on chimney draught, is :

$$H = \left(\frac{0\cdot06\,F}{A}\right)^2$$

A = smallest area in feet after adding 2 ins. round the inside of the chimney to allow for friction losses.

Formulae for the Area of a Chimney.

The area of a chimney depends first of all on the height of the chimney, and then the volume of the gases to be dealt with, that is the amount of coal to be burnt.

With reasonably good brickwork it is necessary to allow 300 cub. ft. of cold air at 32° F. per lb. of coal for natural draught. Now, when the grate area is known and also the amount of coal to be burnt per sq. ft. of grate area, and allowing the above volume of air, it is easy to calculate the volume of air per minute passing up the chimney. This volume of cold air must then be calculated for the temperature which will result in the chimney base.

As already seen, the temperature in a brick chimney under ordinary conditions is reduced about 2° F. for every 3 ft. in height. That is, at a base temperature T*b*, the temperature at the top of the chimney is—

$$Tt = Tb - \frac{H}{3} \times 2 \qquad H = \text{height in feet;}$$

and the average mean temperature in the whole chimney is—

$$T_M = \frac{Tb + Tt}{2}$$

or, to save a double calculation,

$$T_M = tb - \frac{H}{3}$$

Very many formulae have been used for the areas as follows—

where A = inside area at top in sq. ft.
B = weight of air allowed per lb. coal;
D = inside diameter at top in feet;
F = weight of coal burnt per hour in lbs;
G = total grate area in sq. ft.;
H = height of chimney in feet above the grate;
K = for bituminous coal $\frac{F}{1{\cdot}89\,G}$
For anthracite $\frac{F}{G}$
t = temperature in ° F. of flue gases.

(1) Christie : $A = \frac{F}{K\sqrt{H}}$

(2) Gale : $A = 0{\cdot}07\,F_3$

(3) Lange : $A = 0{\cdot}00049$ B.F.

(4) Molesworth : $A = \frac{F}{12\sqrt{H}}$

(5) Smith : $A = \frac{0{\cdot}0825\,F}{\sqrt{H}}$

(6) Kent : $A = \frac{0{\cdot}06\,F}{\sqrt{H}}$

(7) Other formulae : $A = \frac{0{\cdot}061 \times F}{\sqrt{H}}$ (based on rate of combustion of 24 lbs. per sq. foot grate area.

(8) „ $A = \frac{1{\cdot}49 \times G}{\sqrt{H}}$ (with a number of boilers into one chimney, the constant 1·49 can be reduced to 1·20).

To calculate the area of the chimney from first principles for, say, four "Lancashire" boilers, each of 36·00 sq. ft. grate area, coal burnt at 25 lbs. per sq. ft. grate per hour, and temperature in chimney base of 480° F., taking the height as 170 ft. as in the previous example.

The mean temperature of the chimney is—

$$\begin{aligned} T_M &= t_b - \frac{H}{3} \\ &= 480 - \frac{170}{3} \\ &= 480 - 56{\cdot}6 \\ &= 423^\circ \text{ F. say.} \end{aligned}$$

On four boilers at 36·00 sq. ft. grate area per boiler, the total grate area is 144·00 sq. ft. At 25·0 lbs. coal per sq. foot grate area per hour, the total coal burned on the plant per hour is 144·0 × 25·0 = 3,600 lbs. This corresponds to 60·0 lbs. of coal per minute.

Since 300 cub. ft. of cold air per min. is required per lb. of coal, this corresponds to 300 × 60 = 18,000 cub. ft. of air per min.

Since the main temperature in the chimney is 423° F., the real volume passing up the chimney is considerably more than 18,000 cub. ft. per min., in the ratio of the absolute temperature, namely,

$$18{,}000 \times \frac{423 + 461}{32 + 461} = 32{,}210 \text{ cub. ft. per min.}$$

We have now to calculate the velocity of the gases. This velocity is roughly proportional to the square root of the water gauge, and would be exactly so if it was not for the friction of the gases in the chimney. A useful formula is—

$$V = 100\sqrt{H}, \text{ where } H = \text{height in feet.}$$

Thus in the above case,

$$\begin{aligned} V &= 100\sqrt{170} \\ &= 100 \times 13{\cdot}05 \\ &= 1{,}305 \text{ ft. per min.} \end{aligned}$$

Since the volume of the gases is 32,210 cub. ft. per min. and the velocity 1,305 ft. per min., the effective area will be—

$$\frac{32{,}210}{1{,}305} = 24{\cdot}68 \text{ sq. ft.}$$

With a circular chimney, this is approximately 5 ins. 7 ft. diameter and, adding on 4 ins. for the 2 ins. of "dead area," this corresponds to 5 ft. 11 ins., say, 6 ft. diameter.

To Calculate the Approximate Water Gauge.

The intensity of draught of a chimney is expressed as so many "inches water gauge" as obtained in a "U" tube filled with water.

The static pressure in lbs. per sq. ft. is obtained as follows—

$P = Hd_1 - Hd_2$ $\quad$ P = static pressure in lbs. per sq. ft.

$= (d_1 - d_2)$ $\quad$ H = height of chimney in ft.

d_1 = density of outside air ;

d_2 = density of air inside chimney (flue gases).

With H as 170 ft., mean chimney temperature of 423° F., and temperature of air outside 50° F., air at 32° F. has a density of 0·0807 lbs. per cub. ft. At 50° F. density is less in proportion of 0·0807 lbs. per cubic foot.

$$\text{At } 50^\circ \text{ F. density is less in proportion} = \frac{0\cdot0807 \times 32 + 461}{50 + 461}$$

$$= 0\cdot0778 \text{ lbs.}$$

$$\text{at } 423^\circ \text{ F. density, is less in proportion} = \frac{0\cdot0807 \times 32 + 461}{423 + 461}$$

$$= 0\cdot0450 \text{ lbs.}$$

$$\begin{aligned} \text{Therefore } P &= H(d_1 - d_2) \\ &= 170\,(0\cdot0778 - 0\cdot0450) \\ &= 170\,(\times\cdot0328) \\ &= 5\cdot576 \text{ lbs.} \end{aligned}$$

Now it can be calculated that 1 in. suction is equal to 5·2 lbs. per sq. ft., so that the actual water gauge will be—

$$\frac{5\cdot576}{5\cdot20} = 1\cdot07 \text{ in. W.G.}$$

In practice, the actual water gauge will be slightly less, because as soon as the gases are in motion, it tends to fall apart from varying weather conditions.

To find the height of a chimney to give a specific draught—

$$H = \frac{d}{\dfrac{7\cdot6}{t_1} - \dfrac{7\cdot9}{t_2}}$$

H = height in feet ;

d = draught in inches water gauge ;

t_1 = absolute temp. of air ;

t_2 = absolute temp. of flue gases.

CONSTRUCTION OF CHIMNEYS

A chimney may be constructed of brick, stone, ferro-concrete, or steel.

BRICK OR STONE CHIMNEYS.

These are built circular, square, or octagonal ; but there is no doubt that a circular chimney is best, because of its neat appearance and least resistance to wind. Ferro-concrete and steel chimneys are always circular. A good combination is a circular chimney with a square base, not less than, say, 12 ins. high, but depending on the height and weight of the chimney.

The building of chimneys is a specialized branch of the building trade, and the actual construction is more a question for architects and builders than for engineers. The following points are, however, of interest—

The weight of a chimney is very great, and consequently much care has to be taken with the concrete foundations. These should be allowed a number of weeks to set, and the size depends on the weight of the chimney and the nature of the ground. The area of the foundations is usually arranged for average conditions, so that the pressure does not exceed 4 tons per sq. ft., but this depends chiefly on the ground.

Another opinion is that the weight on the ground should not exceed 2 tons per sq. ft. on ordinary compact sand, gravel, or loam. For rock foundations, this can be much increased.

To calculate the weight of a brick chimney, Hutton's formula may be used.

$$W = \frac{CH2 \times b}{B}$$

W = total weight of shaft in lbs. for safety;

H = height of shaft feet;

B = external breadth of shaft at base in feet;

b = mean external breadth of shaft in feet;

C = co-efficient of wind pressure per sq. ft. This depends on the cross-section of the chimney. Roughly for square chimneys the figure is 56, for octagon chimneys 35, and for round chimneys 28.

Weight of built-brick averages 112 lbs. per cub. ft. (say, 100–130 lbs.).

Mean thickness of a square chimney—

$$\frac{W}{H \times 6} \times 448$$

If hexagonal, × 0·75;
If octagonal, × 0·65;
If circular, × 0·55.

Chimneys have also to be built to resist wind pressure. A general rule for diameter of brick chimneys is to make it one-tenth of the height, with a "batter" or taper, $\frac{1}{16}''$ – $\frac{1}{4}''$ per foot on each side.

The thickness of brickwork of a chimney for average conditions is one brick thick (8–9 ins.) for the top 25 ft., decreasing downwards increase the thickness ½ brick for each 25 ft.

A good chimney should have an inner shaft of firebrick for a considerable way up, so that there is an air space between the inner and outer shafts. These two shafts are "bonded" together at intervals, and the result is a non-conducting layer of air which prevents undue cooling of the chimney and reduction of the draught. In construction, cement mortar is much better than the ordinary lime mortar.

When two or more flues enter a chimney at different sides, a mid-feather wall should be built across the inside of the chimney base to direct the air currents from each flue.

It is always good practice to build an arched entrance in each of the four walls of the chimney base, so that in the future any one of these can be used without cutting the chimney.

Chimneys have to be protected from lightning with copper lighting conductors attached to a broad copper band going down outside the chimney attached with copper nails, "earthed" in damp earth (or water, if possible) to get a good connection.

Steel Chimneys.

Steel chimneys are now being more and more introduced, and the modern steel chimney is entirely self-contained, fitted on a square brick base without unsightly guy ropes.

The steel shaft should be lined completely with firebrick to the top. If not, the cooling effect is very excessive, as steel has a much higher conductivity than brick or stone, and the draught is seriously reduced. A space is usually left between this inner firebrick lining and the outer steel shell.

A steel chimney must be thoroughly well painted inside and outside with a number of coats of red lead and good linseed oil, and should also be painted regularly every two or three years.

The weight of a steel chimney is about ⅛ that of an equivalent brick or stone chimney. Advantages claimed for steel chimneys as against brick or stone chimneys are—

1. Greater strength and safety.
2. Smaller cost of installation.
3. Not porous, and no cold air filters in.

The disadvantages are much shorter life, say, thirty years with care; but much less if the best attention is not given.

Concrete Chimneys.

These are chiefly used in America, the concrete being reinforced (even up to 300 ft. high). The chimney is generally constructed of two parts—lower double shell and single shell above—united at the offset. Inside shell about 4 ins. thick and thickness of outer shell 6–12 ins., depending on the height. Between two shells, space of about 4 ins. air.

Advantages and Disadvantages of Chimney Draught.

A. *Advantages.*

1. Simplicity. Once erected, there is nothing to go wrong, as there are no working parts, and practically no attention is required.
2. Practically no cost of upkeep.
3. Very valuable as a discharge for obnoxious gases, and to take away grit and dirt.

B. *Disadvantages.*

1. Very costly to erect, and many times the cost of mechanical draught with short chimney. Brick or stone chimneys become

particularly expensive, as they approach 200 ft. in height. Generally it is better to build two or three small ones, except, of course, in special cases, such as taking chemical fumes away or removing grits from a "white" paper mill. In calculating the expense of erection, interest on the capital outlay must be included along with 2½ per cent. depreciation. Some very large chimneys have, however, been built. The largest chimney in the world is said to be at Great Falls, Montana, U.S.A. Height 506 ft. and 50 ft. internal diameter at the top, designed to move 4,000,000 cub. ft. of air per min. at 600° F. In Great Britain an enormous chimney is Townsends (of Glasgow), 454 ft. high. In ordinary practice, 300 ft. is a very high chimney.

2. Badly effected by climatic conditions and impossible to obtain the intense draught necessary for the highest efficiency.

3. Very low efficiency. As already pointed out, the draught depends on the heat wasted in the chimney base; and to get a good draught, 15–20 per cent. of the coal bill of the plant has to be wasted for ordinary conditions, which is a shocking loss in coal per annum. If it is tried to prevent this loss by thicker fires and complete installation of economizers, the draught is so impeded as to be impossible in practice.

These difficulties are increased by the cooling action of the air on the outside of the chimney, and for this reason it is impracticable to build a very large chimney to increase the draught.

For example, in a chimney 300 ft. high and a temperature 600° F. at the base, the approximate mean temperature in the whole body of the chimney is—

$$\begin{aligned} T_B &= T_M - \frac{300}{3} \\ &= 600^\circ \text{ F.} - 100 \\ &= 500^\circ \text{ F.} \end{aligned}$$

A chimney 400 ft. high would in the same way give a mean temperature of 467° F., and at 500 ft. the mean temperature would only be 434° F.

MECHANICAL DRAUGHT

As already explained (p. 672), chimney draught is unscientific, as the draught depends on the amount of heat wasted in the chimney base. With mechanical draught, this difficulty is overcome, and the draught obtained is entirely independent of the temperature of the flue gases which, theoretically, could be completely cold.

Mechanical draught is applied in three ways, namely, induced draught, forced draught, and balanced draught.

INDUCED DRAUGHT.

This consists of an "inducing" fan, which pulls the products of combustion through the flues and fan, and discharges them into the chimney.

Induced draught is much more commonly used than forced or balanced draught, and is less expensive to apply and simpler in operation.

The fan is fitted with ring lubricated bearings, and the bearing nearest the hot fan is water-cooled. The flow of cooling water is about 3–5 galls. per hour, and troubles due to hot bearings are now a thing of the past on an ordinary quality job.

The fan can be driven by means of a forced lubrication, non-condensing, high speed, enclosed steam engine direct coupled on a combined cast-iron bed plate, and the speed of the fan and engine controlled direct from the firehole. Such an arrangement is very satisfactory in practice. The engine can be condensing, or the exhaust steam fused to heat the feed-water or attached to the condenser of some neighbouring engine. The speed of such a set will be from, say, 250–550 r.p.m., depending on the size of the job. The draught obtained may be anything up to 3–3½ ins. suction water gauge in the fan inlet, according to the specification.

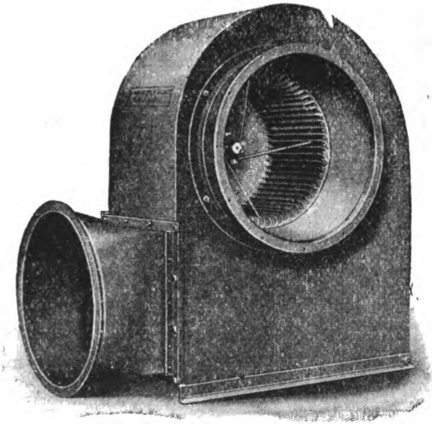


Fig. 25.

The engine can be replaced by a direct motor drive, and the speed will then be, say, 450–750 revs., controlled from the firehole by a "tramway" starter. The fan, of course, can also be driven by belt or chain drive from an ordinary slow speed horizontal steam engine.

It is also possible to drive the fan with belt or chain from a line shaft, although in general this is not good practice, as the great value of the control of the draught is lost, except such regulation as is possible by damper manipulation.

The most modern type of fan is the "multi-bladed," such as the "Davidson" and "Keith-Blackman" fans. Fig. 25 illustrates the "Davidson."

THE "PRAT" SYSTEM OF MECHANICAL DRAUGHT—

. . . as supplied by *M. Louis Prat, 29 Rue de l' Arcada, Paris.* (Representative in Great Britain, Mr. S. Utting, A.M.I.Mech.E., A.M.I.E.E., 82 Victoria Street, Westminster, London, S.W.1.)

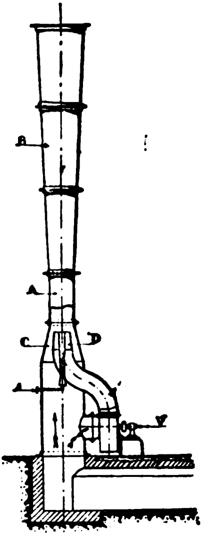


FIG. 26.

The "Prat" system is a special variety of induced draught depending on the principle of draught by induction, the fan itself playing a minor part in the suction of the gases, as shown by the diagrams.

Fig. 26 shows a standard arrangement of the "Prat" system. The steel chimney is comparable to an injector, of which C is the convergent, A the contraction or chamber, and B the diffuser. The flue gas is forced up the chimney through the nozzle D. If the fan is stopped, an emergency steam nozzle C can be used. The mechanical efficiency of the fan used is stated never to fall below 60 per cent., and the bearings are ring lubricated and cooled with air currents, which are claimed to be an improvement on the ordinary water-cooled bearings.

FORCED DRAUGHT.

A forced draught installation consists of a fan which forces air under the fires, with a closed ashpit or into a closed stokehold, as in marine practice. The fan and drive are on the same general lines as for induced draught, except that, of course, water-cooled bearings are not necessary, as air is conveyed to the boiler furnaces through metal trunking, and the firedoors are fitted with an automatic arrangement which shuts off the air blast when the doors are opened to fire. Otherwise the fire would tend to be blown out of the front of the boiler. This system is said to be of special advantage in burning inferior coal.

James Howden & Co., Ltd., Glasgow.—This firm supplies a special type of forced draught, in which the cold air supplied by the forced draught fan, instead of going direct to the boilers as in the ordinary system, is passed through a nest of steel tubes in the boiler side flues and the air heated up to a high temperature, say, 300° F. This hot air is then blown under the fires with the usual type of closed ashpit.

BALANCED DRAUGHT.

Balanced draught is a combination of forced and induced draught, in which one fan acts as an induced draught fan and a second as a forced draught fan on the same fires, and consequently the draught is "balanced," that is there is neither suction nor pressure above the fire.

The induced draught is produced by a suction fan, which is placed at the base of the chimney and, drawing the waste products of combustion through the vertical tubes of the air-heating economizers, delivers them into the main flue or chimney.

The forced draught is produced by a pressure fan, which delivers the correct quantity of air required for combustion into the economizer.

The term "Balanced Draught" is used because the fans are so proportioned that the vacuum set up by the suction fan is just sufficient to overcome the resistance due to the flues, tubes, etc., when the pressure fan is setting up the requisite under-grate pressure to burn the required amount of coal.

The draught at the furnace door is thus balanced, so that there is no outrush of flame or inrush of cold air when the furnace doors are opened, as is the case when forced or induced draught is used separately.

THE ADVANTAGES OF MECHANICAL DRAUGHT.

The advantages claimed by mechanical draught are—

1. Much less initial cost than a high chimney, which is rendered unnecessary.
2. Draught independent of atmospheric conditions.
3. Draught can be varied at will to suit the quality of coal and the varying demands for steam.
4. Full use can be made of economizers, and the flue gases cooled down to any extent so as to make the best advantage of the economizers.
5. Increases the evaporation of the boiler plant.
6. Gives a greatly increased efficiency, because the grates can be shortened and a thick white-hot fire obtained with a high percentage of CO_2. Only takes 250–300 cub. ft. of air per lb. coal.
7. Enables inferior coal to be burnt economically, which is impossible with natural draught.
8. Gives a very greatly increased rate of combustion—25–40 lbs. of coal per sq. ft. grate area per hour, depending on the quality of the coal.
9. Also can, if necessary, use a very hot air supply, up to, say, 300° F., if waste heat is available.

DISADVANTAGES OF MECHANICAL DRAUGHT.

1. Power taken to drive the fan on good installation is

1–2½ per cent. of the steam production of the plant, whilst on bad installations it may be greater.

2. Requires more skilled attention, and is liable to give worse results than chimney draught if not in competent hands.

3. Being a machine, is liable to breakdown and wear and tear. Average life with good attention can be taken as fifteen years.

To Calculate the Power required for Fans.

Taking a motor-driven fan.

$$\text{Kw. required} = \frac{DAV^3}{2{,}454{,}000}$$

D = density of flue gases in lbs. per cubic foot;

A = area of discharge opening in square inches;

V = velocity of gases in feet per second, that is periphy speed of fan blades.

This assumes 55% efficiency fan,
„ 92% „ motor.

Another formula,

$$\text{B.H.P.} = \frac{W \times C \times A \times P \times 5{\cdot}2}{60 + 33{,}000} \times \frac{100}{E}$$

W = average weight of coal consumed per hour;

C = cubic feet of air per lb. coal (generally 300 cub. ft.);

A = coefficient of volume of gas at flue gas temp. (T) as compared with vol. of air at 50° F. (T_1) calculated as absolute $\frac{461 + T}{461 + T_1}$

P = draught in inches W.G. at fan pressure or ;

$\frac{100}{E}$ = efficiency of fan (52–55%).

5·2 = constant for converting pressure or draught as read in inches W.G. to pressure in lbs. per sq. ft.

Taking 52–55% fan efficiency and 300 cub. ft. of air, we have:

$$\text{H.P.} = \frac{W \times A \times P}{660}$$

This is actual B.H.P. to drive the fan, and efficiency of the motor has to be added.

On large plants, figure is roughly 0·55 B.H.P. per 1,000 lbs. water evaporated per hour; or, say, 1–2½ per cent. of the steam production of the plant.

Fire-bars.

The ordinary fire-bar for hand-firing on all types of boiler is of fairly standard type although there are numerous modifications. The dimensions of the bar depend on the personal opinion of the user. The total length of the grate may be anything from 5′ to 7′ 6″. An average size is 6′, that is, two short bars of 3′ each. The width of the bar may be ½″–1½″, and the air space ⅛″–½″. A bar which is too slender tends to warp and burn out, whilst a bar too heavy restricts the air space.

Special Fire-bars.

A special type of bar, out of the ordinary, is *Whittaker's Water-cooled Fire-bars*, as supplied by Messrs. John J. Whittaker, Ltd., Burnley Road, Accrington. These bars consist of an arrangement in which each bar has a water trough. These troughs are connected with each other and kept supplied with water. By this means, water is evaporated under the whole surface of the great area of the furnace, with the object chiefly of preventing clinkering.

Another general type of bar is the rocking bar, in which the bars can be rocked to and from at intervals, either mechanically or by hand. The object of this is to loosen the clinker, and shake the ashes through to keep a cleaner fire and reduce the amount of cleaning out. This is an obvious advantage. The disadvantage is that the bars have to be very strong, and consequently the air space is apt to be restricted, and also the men often will not rock them because of the extra work. Typical of this type of bar, we have—

"*Knowles's*" *Rocking Bar*, as supplied by Messrs. Knowles & Co., Globe Works, Toller Lane, Bradford, Yorks. These bars are made of chilled metal, and rock across the grate parallel to the front of the boiler, so that the bars are very short and strong. The bars are actuated by a lever under the front of the furnace, and each alternate bar moves.

Neill's Rocking Bar, as supplied by Messrs. James Neill, Temple Park Ironworks, Glasgow. These bars rock parallel with the furnace, in two groups of, say, 3 ft. each group; but rock as one unbroken bar actuated by level under the front of the furnace at the side; made of chilled metal; and bars can be adopted to be rocked mechanically, if necessary, as supplied with an economic "split" bridge with a view to smoke prevention.

"*Yates & Thom*" *Rocking Bar*, as supplied by Messrs. Yates & Thom, Ltd., Canal Engineering Works, Blackburn. These bars also rock across the grate; are semicircular in form; and, when tilted, present a quarter circle above the grate level, actuated by a lever under the front of the furnace, the handle of which can be turned on one side when not in use.

Special Steam Jet Furnaces.

The general principle of these furnaces is that of a hollow fire with steam jets blowing through the hollow space, so that a mixture of steam and air is forced through the fires above. In many cases the steam is "superheated" by passing the inlet pipe through the side flues of the boiler before arriving at the nozzles. It is generally claimed by the makers that this arrangement gives increased efficiency, increased evaporation, less smoke, increased facilities for burning inferior fuels, greatly increased life of the bars, prevention of clinkering, very quick steam-raising, etc. The efficiency of these furnaces is a matter of opinion, and the quality of fuel used and the method of working are important factors.

There is a very great difference of opinion as to the amount of steam taken by these furnaces. Most of the makers state figures of, say, 1–3 per cent. of the production of the plant; and, in many text-books of engineering, the figure is stated as about 5 per cent. The United States Navy Department (1890) published data in which the figures were stated to be 8·3–21·2 per cent. of the production. The Author has studied this question for some years, and in *Engineering*, 16th January, 1920 ("Exact Data on the Running of Steam Boiler Plants: No. 3—the Amount of Steam used by Steam-Jets"), gives the results of investigations on eleven different types of steam-jet furnaces. These investigations were carried out on fifty-four different boiler plants, representing 149 boilers, with a coal bill of about 312,000 tons of coal per annum.

It is not an easy matter to determine the amount of steam issuing from a nozzle, and the method employed was that of condensing the steam by external cooling and weighing of the condensed water. The Author has devised a special apparatus for the purpose, consisting of an enclosed horizontal cylinder connected with a long vertical condensing coil. The cylinder is jacketed, and cold water is run through the jacket so as to cool the interior. The vertical coil is also immersed in a large volume of cooling water. In determining the steam used by any particular system of steam jets, our apparatus is erected in the fire-hole close up against the boiler. The steam pipe and jet apparatus is then taken out of the boiler furnace and placed bodily inside the horizontal cylinder and coupled up in exactly the same manner as when in the boiler furnace. The end of the horizontal cylinder is closed, and on opening the steam supply valve the steam-jet apparatus works in the enclosed cylinder for all practical purposes exactly the same as it does in the furnaces.

Care was taken that the supply valve was open to the same extent as when used under the furnaces, and that the steam pressure on the jets was also as usual. The steam issuing from the jets is condensed in the horizontal cylinder and the condensing coil, and runs out of the end of the coil as cold condensed water, and is weighed direct. There is a very minute pressure in the cylinder, caused by the small resistance to the passage of the condensed water through the coil, and this will tend, therefore, to make the figure for the steam used by the jets very slightly lower than would be the case in the open air in the furnaces; and this slight error, if appreciable, is therefore on the right side. In this way, a series of observations were carried out, say, three or four successive determinations of one half-hour each; and if the figures for the several determinations were similar, this was sufficient. If there was much difference, a further series of determinations was carried out to get good average figures. A fair amount of judgment is, of course, necessary in these determinations. In the first place, there is the question of the amount of opening of the steam supply valve to the jets. On a ½ in. or ¾-in. pipe, for example, say, three-quarter turn is generally the same as full-open as regards the amount of steam passed; but in 9 per cent. of cases, the firemen simply open the valve full open, that is, two or three turns, and leave it all day continuously in this position. In such a case, the method of determination is simple, the valve being full open. In some cases, however, the firemen are more careful and work with, say, quarter turn open on an eight-hour shift for, say, six hours, and full open for, say, two hours, when cleaning out or when the load is particularly heavy. In such a case, the practice of the firemen was carefully observed, and separate determinations were carried out accordingly with the valve quarter-turn open and full open. The two results were then averaged up in the proportion of six hours to two hours, and in this way a good average determination was secured.

Secondly, there is the question of variation in the steam pressure. This presents no difficulty in the case of most boiler plants where the steam pressure is kept up to within a reasonable margin of, say, 20 lbs. In those cases where the steam

Average Results for Steam-jet Furnaces, Hand-fired

(*Averages* 6·6 *per cent. of the production*)

Type of Apparatus.	No. of Plants.	No. of Boilers.	Average % of Product used by Jets.	Total Coal Bill.	Total Coal Bill used by Jets.
			Per cent.	Tons per ann.	Tons per ann.
Type A ..	6	17	7·6	20,200	1,535·2
„ B ..	4	7	4·5	15,400	693·0
„ C ..	3	6	7·3	9,234	674·0
„ D ..	18	60	6·3	137,005	8,631·5
„ E ..	7	22	8·1	55,450	4,491·4
„ F ..	2	3	3·2	5,850	187·2
„ G ..	2	4	5·0	6,600	330·0
„ H ..	2	6	7·7	14,400	1,108·8
„ I ..	3	6	4·4	8,525	375·1
„ J ..	1	2	15·25	4,000	610·0
„ K ..	6	16	5·9	35,346	2,079·5
Total	54	149		312,010	20,715·7

pressure cannot be maintained within reasonable limits, it is necessary to carry out the determination at various pressures and average the results approximately in the proportion of the pressures being obtained in practice.

There is also the question of the wearing action of the steam on the size of the holes or nozzles through which the steam is blowing. The holes gradually tend to wear larger, depending on the time the apparatus has been in operation, and, therefore, different results may be obtained with the same apparatus depending on this condition. The determinations were carried out exactly as the plants were found working and, because of the large number of plants tested, the conditions in this respect must, as a whole, be regarded as affording an excellent average. The results obtained are tabulated as follows, the average of the whole fifty-four plants being 6·6 per cent. of the production. There is, however, very great differences on individual plants, and the figures vary from 1·0 per cent. to 15·25 per cent. of the production, averaging 6·6 per cent.

It is a difficult matter to estimate the amount of steam issuing from an orifice by means of a formulae. The formulae usually adopted is Rankines', which is as follows—

$$W = \frac{6ap}{7}$$

W = weight in lbs.;
a = area in square inches;
p = absolute pressure.

The theoretical flow has then to be multiplied by—

K = 0·93 for short pipe,
0·63 for thin opening—such as a safety valve.

BOILER FEED WATER

Pure rain water is represented by the chemical formula H_2O, but such pure water is never found in Nature. As rain (pure) water falls through the air, it absorbs minute quantities of impurities, such as dust, sulphurous gases, ammonia, carbon dioxide, and so on. As soon as the rain water reaches the ground and percolates through the soil, it rapidly absorbs a combination of substances depending on the nature of the ground.

In the first place, it dissolves carbon dioxide (CO_2), forming carbonic acid (H_2CO_3) from decomposing vegetable matter, as well as complicated organic acids, known as "peaty" acids. With this impurity present, it then dissolves carbonates of lime and magnesia, forming soluble bicarbonates. In a limestone country, this is the chief cause of the very hard water; and the serious results of this action are well illustrated by the hundreds of caves formed in a limestone district such as Derbyshire by this dissolving action of water containing carbonic acid.

Rain water percolating through the earth also dissolves calcium acid and magnesium sulphates, which are present in many districts as constituents of the soil. In the same way, more or less quantities of other salts are liable to be present, such as sodium, calcium, and magnesium chlorides and nitrates. Further impurities may be carbonic acid and air.

Because of these impurities in natural water, scale is formed in the boilers and, in many cases, corrosion or "pitting" also results. Scale is formed in boilers by two distinct processes, which may go on together, or separately, depending on the quality of the water.

The first process is that due to what is called "temporary" hardness. Calcium carbonate has the formula $CaCO_3$, and is practically insoluble in water (1 pt. in 62,500 at 212° F.). Magnesium carbonate has the formula $MgCO_3$, and is also very insoluble (1 pt. in 9,600 at 212° F.). These two salts are therefore harmless, because they will not dissolve in water in sufficient quantities to cause scale. When, however, carbonic acid is present in water, calcium and magnesium carbonates are dissolved, forming calcium bicarbonate $CaH_2(CO_3)_2$ and magnesium bicarbonate $MgH_2(CO_3)_2$, which are extremely soluble in water.

Very large amounts of calcium and magnesium salts can therefore be present in the water, which are carried into the boilers. Calcium and magnesium bicarbonates are, however, very unstable compounds, and even below 212° F. they are decomposed again, precipitating insoluble calcium and magnesium carbonates, and liberating free CO_2. If, therefore, water containing "temporary" hardness, that is, calcium and magnesium bicarbonates in solution, is merely boiled, all the temporary hardness is precipitated and the water is then soft. Consequently a feed-water containing temporary hardness deposits calcium and magnesium carbonates in the boilers and economizers, forming mud and also hard scale.

The second process is that due to "permanent" hardness, which is caused by calcium sulphate $CaSO_4$. This salt is fairly soluble in water (1 pt. in 460 at 212° F.) and is carried into the boiler plant in solution. As the water in the boiler evaporates and forms steam (pure water), the calcium sulphate gradually accumulates in the boiler, and forms a stronger and stronger solution. It finally becomes "saturated," that is no more calcium sulphate will stop in solution, and it then crystallizes on the boiler plates. This forms the worst type of scale, a hard tenacious deposit, especially where the greatest heat is present, as on the furnace tubes. It will be observed there is no chemical reaction and the calcium sulphate is not decomposed, but simply deposits by crystallization. For this reason, it is called "permanent" hardness, as it is not decomposed by boiling.

Magnesium sulphate ($MgSO_4$) is very soluble in water (33·8 pts. in 100 pts. at 60° F.), and for this reason, as a rule, does not form scale, as the concentration in the boiler is never sufficient for crystallization.

In addition to scale formation, we have also the trouble due to corrosion. In this connection, magnesium salts are most objectionable. Magnesium chloride ($MgCl_2$) is very soluble (55 pts. in 100 at 60° F.), and is extremely dangerous, because it easily decomposes to form magnesium hydrate $(MgOH)_2$ and free hydrochloric acid H.Cl., which at once attacks the boiler plates. $MgCl_2 + 2H_2O = Mg(OH)_2 + 2HCl$.

Magnesium hydrate also forms a very bad scale. The danger with magnesium salts of any kind is that they are apt to react with other salts to form the dangerous magnesium chloride. Thus magnesium sulphate ($MgSO_4$) will interact with calcium chloride (which is otherwise harmless) to form magnesium chloride. Also with sodium chloride (common salt). Magnesium sulphate will also react with calcium bicarbonate to give magnesium hydrate and calcium sulphate, both of which form a very bad scale. Sodium sulphate in water is harmless. Sodium chloride is also harmless in ordinary quantities, except when magnesium salts are present. In large quantities it may cause corrosion by electrolytic action, in which the boiler acts as an electric cell.

Scale is a very serious matter in a boiler plant. A deposit of scale on a boiler plate or tube reduces the conductivity at once, and not only causes a drop in the efficiency of the boiler plant, but also reduces the plate or tube liable to drainage by heat. This is particularly serious in water-tube boilers.

According to Rankine, the resistance of carbonate of calcium is 16–18 times that of iron, and sulphate of calcium 45–50 times that of iron. He calculated $\frac{1}{4}$ in. of scale would increase the fuel bill 50 per cent.

Stromeyer, however, says that with a "Lancashire" boiler and economizers, the drop in efficiency is not more than 5 per cent. with, say, $\frac{1}{4}$ in. of scale; but that in water-tube boilers, even a good feed-water soon begins to drop the efficiency (i.e. 1,000-hr. run will drop the efficiency 10 per cent.).

Other authorities state that $\frac{1}{64}$ in. of scale corresponds to 2 per cent. loss in the coal bill, $\frac{1}{32}$ in. to 4 per cent. loss, and $\frac{1}{16}$ in. to 9–10 per cent. loss. It is also stated that $\frac{1}{16}$ in. of scale requires 16 per cent. more fuel to transmit the same amount of heat: $\frac{1}{4}$ in. scale requires 50 per cent. more fuel, $\frac{1}{2}$ in. scale requires 150 per cent. more.

With regard to the effect of the heat of the furnace on a boiler plate or tube covered with non-conducting scale, so that the heat cannot get away freely to the water in the boiler, Stromeyer says that if the water in a boiler is, say, 380° F., the boiler plate is only about 400° F. when no scale is in the boiler, so that the heat is transmitted at once to the water. He states, further, that if $\frac{1}{8}$ in. of scale is deposited, the temperature of the plate goes up to about 700° F., and increased wear and tear and strain on the plates. The presence of oil in addition to scale, increases this bad effect.

Analysis of Boiler Feed-water.

From an engineering point of view, a simple and effective method is that of the soap test: 70 cc. of the water is placed in a glass bottle, and there is added from a burette a standard soap solution, in which 1 cc. of the solution is equal to 1 grain per gallon hardness. This standard soap solution is made by dissolving 13 grs. of Castile soap in 500 cc. of methylated spirits and made up to 1,000 cc. with water. It can be tested by using a standard solution of calcium chloride. This soap solution is added from the burette 1 cc. at a time, and the bottle is closed and violently shaken. As soon as the froth disappears, another cc. is added, and so on until a permanent lather or froth is left of standing 5 mins. The number of cubic centimetres in the burette minus one will be the total hardness, temporary and permanent, in grains per gallon. Thus 15 cc. soap solution is 14 grs. per gallon total. The reason of subtracting one is because it takes 1 cc. even with pure distilled water to give a lather.

Another 70 cc. is now taken and boiled for a few minutes, cooled, and filled up to 70 cc. again with distilled water. This is now titrated as before, when the number of cubic centimetres gives the permanent hardness (i.e. 10 cc. equals 9 grs. per gall. permanent hardness). Since the total was 14 grs., the temporary hardness by difference is 5 grs.

Used with discretion, quite good results are obtained by the soap test in this manner. To get a proper lather, no more than 16 or 17 cc. of soap solution should be used. If the water is very hard, it is best to take, say, 35 cc., add 35 cc. distilled water, and multiply by 2.

The hardness of boiler feed-water may be anything from practically nil to 100 grs. per gallon. Drinking water of towns like Manchester and Glasgow, with exceptionally good water, is only 4–5 grs. per gallon.

The amount of scale deposited in boilers is enormous. The Author gave in *Engineering*, 12–19th July, 1918 ("Coal-saving by the Scientific Control of Steam Boiler Plants"), the results of an investigation of 250 typical boiler plants, comprising 1,000 boilers, with an annual coal bill of 2,166,000 tons.

The average hardness was about 12 grs. per gallon total hardness. Out of the 250 plants, 75 plants had good water, 155 were "medium" with a considerable amount of scale, and 20 cases were "chronic." In only 43 plants was a softening plant in use, and 25 plants were using boiler compositions.

Taking the average hardness of boiler feed-water for Great Britain as 12 grs. per gallon total, by calculation from the approximate amount of coal used for steam generation, this amounts to a weekly evaporation of about 2,128,000,000 galls. to 2,845,000,000 galls.: this would mean that all the boiler plants of Great Britain have deposited in them weekly no less than 1,600 tons to 2,200 tons of solid material, most of which forms scale.

WATER SOFTENING.

The general principle of water softening is as follows—

1. "*Temporary*" *hardness*, consisting of soluble bicarbonates of lime and magnesia. To the water is added the correct amount of slaked lime to combine with the bicarbonate to form only insoluble calcium and magnesium carbonate. This is then reduced by filtration.

$$CaH_2(CO_3)_2 + Ca(OH)_2 = 2CaCO_3 + 2H_2O.$$

With bicarbonate of magnesium, the effect is a little complicated by the fact that the precipitated carbonate of magnesia is slightly soluble in water, and not so easily removed by filtration as the calcium carbonate.

With the correct amount of lime, the chemical reaction is—

$$MgH_2(CO_3)_2 + Ca(OH)_2 = MgCO_3 + CaCO_3 + H_2O.$$

If excess lime is used, magnesium hydrate is formed, which is much more insoluble.

$$MgCO_3 + Ca(OH)_2 = Mg(OH)_2 + CaCO_3.$$

2. "*Permanent*" *hardness*, consisting of sulphate of lime (and magnesia) and, to a small extent, chlorides and nitrates. To the water is added the correct amount of carbonate of soda Na_2CO_3 (soda ash or alkali), which forms insoluble calcium and magnesium sulphates, which are precipitated and removed by filtration. The sodium sulphate left in the water is harmless.

$$CaSO_4 + Na_2CO_3 = CaCO_3 + Na_2SO_4$$
$$MgSO_4 + Na_2CO_3 = MgCO_3 + NaSO_4$$

Also if free carbonic acid (CO_2) is present, this is converted into soluble bicarbonate, which is removed by the lime treatment already described.

$$CaSO_4 + Na_2CO_3 + H_2CO_3 = CaH_2(CO_3)_2 + Na_2SO_4.$$

The first process of water softening on a large scale was that devised by Dr. Clerk, who used lime to remove CO_2 and bicarbonate; then Dr. Porter invented the sodium carbonate treatment to remove sulphates, and the combined process is the original "Porter-Clerk" process.

In this process the hard water is run into huge tanks, and the calculated amount of lime and sodium carbonate added. The tank is stirred up by compressed air or other means, and the reaction takes place at once. The contents of the tank are then allowed to stand for 24 hrs. for the finely divided precipitate of $CaCO_3$ to settle, and the clear softened water is run off, the tank refilled with hard water, and so on. This process is still at work in a few factories in Great Britain. The trouble is that the precipitated $CaCO_3$ takes 24 hrs. to settle. Consequently, enormous tanks have to be used, at least two, each capable of holding 24 hrs. supply.

The modern type of softening plant is a development of the "Porter-Clerk" process, in which improved methods are adopted for quickly separating the precipitated calcium carbonate, so that the process is continuous; and very much smaller tanks can be used, together also with improved devices for measuring and adding the lime and soda.

In a good style of modern plant, say, 3 hrs. settling only is required for the best result instead of the original 24 hrs. A cheap form of modern plant only allows 2 hrs. settling, or even 1 hr., but this is not satisfactory.

Softening plants can also be conveniently divided into two groups, according as to whether they use the lime in the form of lime cream, a mixture of lime and water of the consistency of cream; or in the form of clear lime water, using a much larger bulk of water to make the solution.

All these plants work on the same general principle. The hard water is pumped into the top of the plant, and the flow of this hard water works a small water-wheel or similar apparatus that drives the mechanism of the whole plant. The soda is mixed in one tank and the lime in another (either as cream or lime water), using hard water; and by means of suitable mechanism a small amount of soda solution and of lime is added to the hard water as it falls down into the body of the plant. The relative amounts of soda and lime to the bulk of the water are controlled by adjusting the mechanism depending on the quality of the hard water as found by analysis. Once this is adjusted, the plant is automatic, because the speed of the addition of the two chemicals is controlled by the addition of the hard water, and this admission is again controlled by a float in the softened water reservoir. Thus as the demand for softened water falls off, the float rises and shuts off the inlet hard water, which causes the driving mechanism to slow down, and vice versa.

The hard water falling down the plant is mixed by an agitator with the lime and soda, and the water is softened with the precipitation of calcium carbonate ($CaCO_3$), as already explained on page 692. The water with the precipitate now fills up the settling tank or main body of the softening plant, and most of the precipitate settles down on the bottom of the tank, and is removed at intervals by a sludge pipe. The water then passes through a filter of wood wool, which separates the last trace of precipitate, and into a reservoir for softened water, from which it is taken as required.

There is considerable differences in detail between the plants of the various makers, particularly in the method of adding the soda and the lime, the method of sludging, and the method of cleaning the filters. Illustrations of typical Harris-Anderson, and Kennicott plants are given, and for further information the makers must be consulted direct.

The plants can be arranged either vertical or horizontal, depending on circumstances, and there is little or no difference in efficiency between the two. Also the water entering the plants can be heated practically to boiling by using any exhaust steam available, although the plants work perfectly well with cold water.

Unless there is great variation in the quality of the hard water, a softening plant worked on correct lines will reduce the hardness to, say, 5–6° total hardness, and the water should be analysed at least once a day by the soap test.

An unusual system of water purification is that of the "*Permutit,*" as supplied by Messrs. The United Water Softeners, Ltd., Imperial House, Kingsway, London, W.C.

The "Permutit" process consists in running the hard water through a layer of "permutit," which is a prepared zeolite.

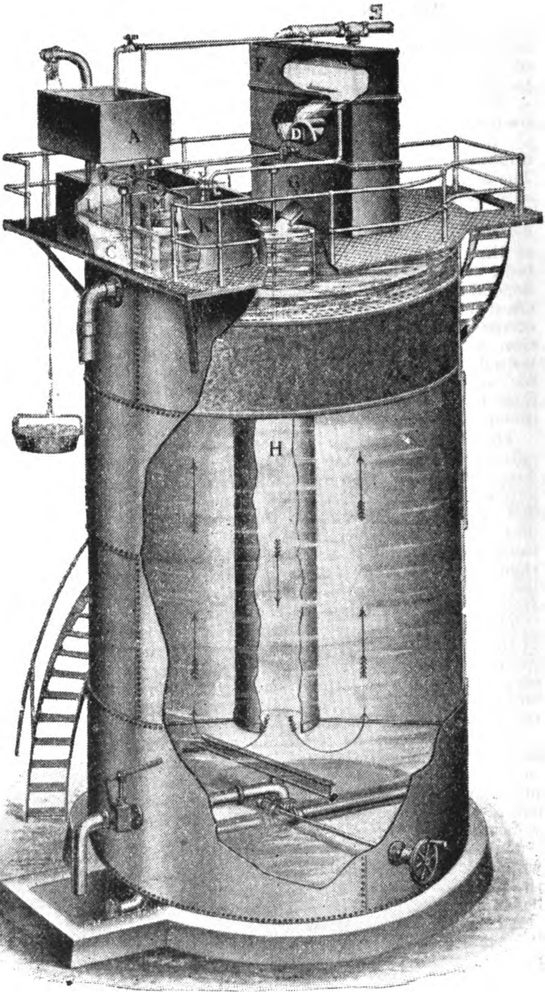


FIG. 27.—"KENNICOTT" WATER SOFTENING PLANT

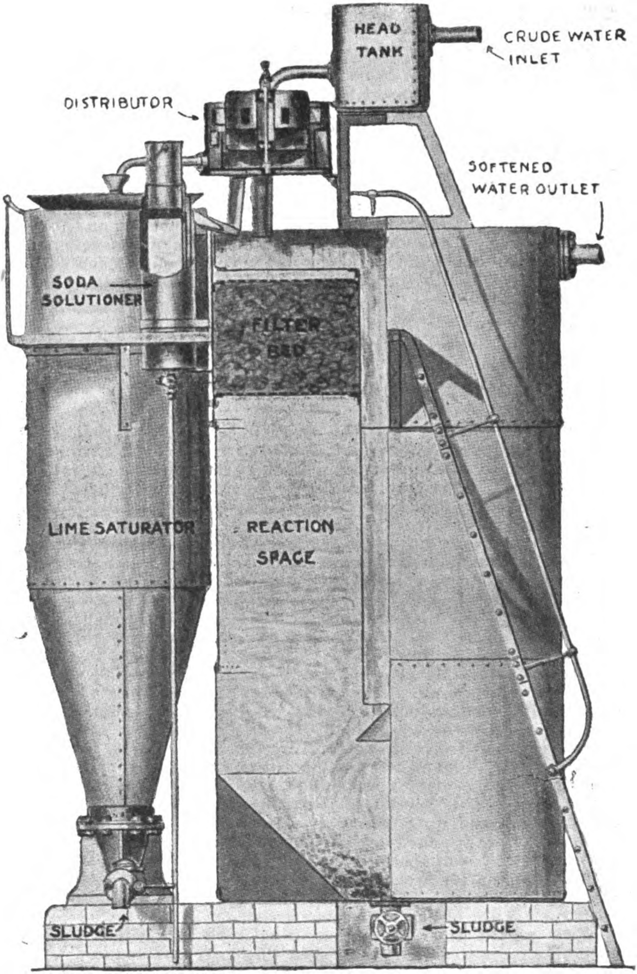


Fig. 28.—Sectional Elevation of "Harris-Anderson" Patent Automatic Water Softener, with Internal Filter.

23—(5016)

A "zeolite" is a mineral substance of varying combinations of aluminium, and the bases of the alkalis (Na, K., etc.) and alkaline earths (CaMg, etc.) with silica acid, the whole containing water of hydration. Many of them possess the property of exchanging a greater or smaller part of their metallic bases for other metals with which they may be brought into contact. The "zeolite" used is of artificial manufacture, specially for "exchanging" with the calcium and magnesium salts of hard water. It is aluminium sodium silicate.

Any calcium or magnesium salts in solution interact at once with the "zeolite," leaving behind insoluble aluminium calcium, whilst the sodium of the zeolite forms the corresponding sodium salt which goes into solution. Thus with calcium salts, the chemical reaction is—

Temporary,

$$\mathrm{CaH_2(CO_3)2} + \mathrm{2Na}\,\chi = \mathrm{NaCO_3} + \mathrm{Ca}\,\chi.$$

Permanent,

$$\mathrm{CaSO_4} + \mathrm{2Na}\,\chi = \mathrm{Na_2SO_4} + \mathrm{Ca}\,\chi\,;$$

$$\chi = \text{zeolite base}$$

and with magnesium salts the reaction is the same, that is, all the temporary hardness is left behind as insoluble magnesium, zeolite, and sodium carbonate passes on the equivalent into the boiler. In the same way, all the permanent hardness is left behind and sodium sulphate passes into solution.

The action is instantaneous, hard water running in at the top, and soft water with 0° of hardness, but containing sodium sulphate and sodium carbonate, running out at the bottom. This action goes on continuously until all the sodium zeolite in the bed is exhausted. When this happens, the process is stopped and a solution of common salt is poured through the zeolite bed. The CaMg zeolite is then instantaneously reconverted to active sodium zeolite, and the CaMg runs away to waste as Ca and Mg chloride. The softening process is then restarted, and so on. The activity of the zeolite goes on apparently indefinitely so long as it is "revived" with salt. This process can be adopted, by the use of a special zeolite, to get rid of all traces in water of iron and manganese.

The whole "Permutit" plant consists simply of a cylinder containing the zeolite as a granular powder, together with a receptacle to hold the salt solution for regeneration. Thus it can be connected up direct to a pressure main source supply of hard water, which flows on automatically softened.

"Permutit" is of special value in connection with the treatment of water for washing, dyeing, and bleaching, etc., in which soap is used. Thus, roughly, every degree of hardness destroys 1·3 lbs. of soap per 1,000 galls.

With regard to removing existing scale in boilers and economizers, if the water is treated efficiently and reduced to less than 5° hardness, the scale will very slowly begin to re-dissolve and loosen, and after a number of months the boiler will be clean. Chemical mixtures sold to take off scale

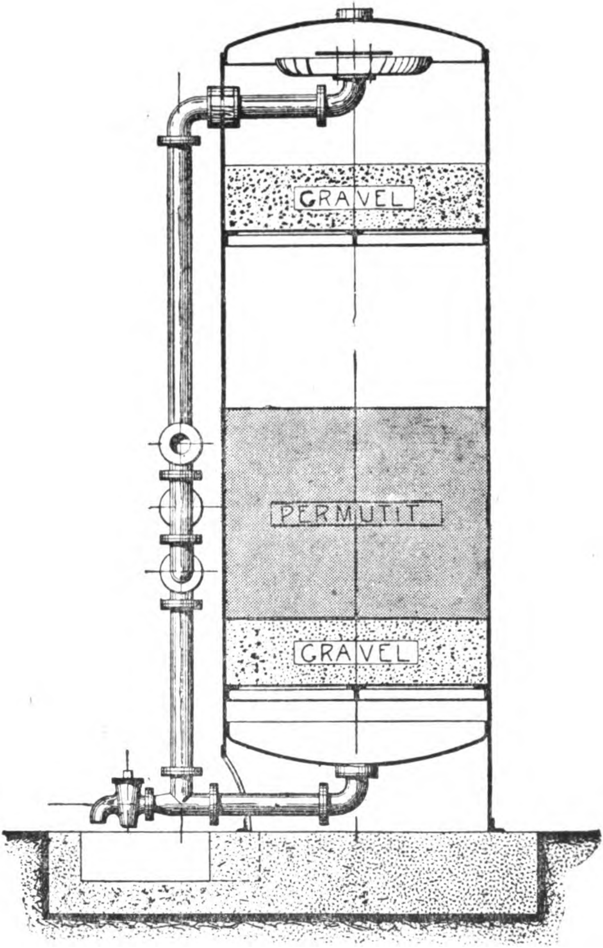


FIG. 27.—"PERMUTIT" WATER PURIFICATION PLANT.

quickly are, in most cases, either dangerous or merely quack remedies. The best way is to clean off all the scale mechanically either by hand or by pneumatic tools, and to start afresh with a clean boiler.

BOILER FEED-WATER TREATMENT WITHOUT A SOFTENING PLANT.

The following processes have been in general use for many years—

1. *Tribasic Phosphate of Soda* (Na_3PO_4), which is made by heating together the correct proportions of ordinary phosphate of soda (dibasic phosphate of soda Na_2HPO_4) and caustic soda, and allowing to crystallize. A solution of this to be added to the boiler feed-water as required, preferably by a small suction pipe attached to the boiler feed pump, so that the treatment is continuous, irrespective of the amount of water pumped.

The effect is that all the objectionable calcium and magnesium salts, sulphates, bicarbonates, and chlorides are precipitated as insoluble calcium and magnesium phosphates in the boiler as a soft mud, which can be blown out as required. The feed-water is left with no hardness at all if the treatment is properly applied.

2. *Sodium Oxalate.* The principle and method of application is on the same general lines, the sodium oxalate forming insoluble calcium and magnesium oxalates.

3. *Tannate of Soda.* This also is on the same general lines, the idea being that tannate of soda forms insoluble calcium and magnesium tannate, which forms an insoluble flocculent precipitate which forms a soft mud in the boiler. The efficiency of tannate of soda in this direction is, however, a matter of opinion. The tannate of soda reacts with the calcium and magnesium bicarbonates and forms carbonate of soda. This then reacts with the calcium and magnesium sulphates, forming magnesium and calcium carbonate, which is insoluble, and leaves sodium sulphate in solution. The CaMg carbonate is apt to bake on the plates to form scale again.

4. *Caustic Soda.* Many engineers are of the opinion that caustic soda is a good preventive of scale. The action is to form sodium carbonate with the calcium and magnesium bicarbonates—

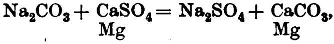
$$2NaOH + \underset{Mg}{Ca}(HCO_3)_2 = \underset{Mg}{Ca}CO_3 + Na_2CO_3$$

leaving a precipitate of insoluble calcium and magnesium carbonate. The sodium carbonate then acts on the calcium and magnesium sulphate, forming as before insoluble Ca and Mg carbonates. The difficulty is that in most cases the right proportions of permanent and temporary hardness are not present for the above chemical reactions to take place to leave the water free from scale-forming elements.

Also the deposited Ca and Mg carbonates are apt to deposit on the plates of the boiler and bake into scale again.

If excess caustic is added, which is very easy, this has a deleterious action on the gun-metal fittings of the boiler, the caustic soda dissolving the zinc out of the gun-metal (zinc tin alloy). Also the action is very severe on packing rings. The Author does not recommend the use of caustic soda. The treatment is also very expensive, as caustic soda costs much more than lime and soda.

5. *Soda Ash.* In some cases, ordinary soda ash (Na_2CO_3) is of some advantage if the water contains nothing but permanent hardness, calcium and magnesium sulphates. The action is, as before—



$$Na_2CO_3 + \underset{Mg}{Ca}SO_4 = Na_2SO_4 + \underset{Mg}{Ca}CO_3,$$

forming insoluble calcium and magnesium carbonates. Soda ash has no action at all on temporary hardness, Ca and Mg carbonates. Here, again, the precipitated CaMg carbonates are apt to bake on the plates to form scale again.

6. *Graphite.* It is claimed that powdered graphite introduced into boiler feed-water will prevent all formation of scale. The action is to keep the Ca and Mg sulphates and carbonates as a fine powder instead of a compact scale, the greasy particles of graphite preventing the particles adhering together to form a scale. It is also claimed that graphite will gradually fetch off existing scale by loosening it.

BOILER COMPOSITIONS.

A very large number of boiler compositions or boiler fluids are on the market, many of them most expensive in price and claiming the most extraordinary properties in preventing scale.

The Author has analysed several hundreds of these compositions, and in most cases they are practically worthless and merely quack remedies.

The great majority of these compositions appear to be made by boiling up a weak solution of residue caustic soda with some cheap and refuse organic matter of vegetable origin. An average figure for composition is, say, 90 per cent. water, 3·5 per cent. caustic soda, 0·5 per cent. carbonate of soda, and 6 per cent. organic matter, mostly of vegetable origin. In most cases, it is almost impossible to trace exactly the composition of the vegetable matter, but spent tan is largely used. The general idea seems to be to make a weak solution of tannate of soda, using the original vegetable matter instead of extracted tannic acid.

Objections to boiler compositions may be summarized as follows—

1. Excess expense, exorbitant prices being charged for ordinary cheap chemicals.
2. Feed-water, except when not more than, say, 6–7°, should be treated before the boiler and all the precipitate separated.
3. Not scientific practice to put unknown compositions into boilers.
4. Affects the steam, generally causing it to become alkaline, which is bad for cooking, brewing, dyeing, etc.

CORROSION.

Corrosion in boilers is generally due to the following reasons, although some cases of corrosion are very mysterious—

1. *Dissolved Gases*, chiefly CO_2 and air. These gases can be in the water at its source, or by absorption from the air in storage tanks. Also may be due to absorbing acid chemical fumes. The corrosive action of oxygen must, however, have CO_2 present to form a soluble salt of iron. The general remedy is to keep the feed-water alkaline.

2. *Acid in the Water* may be due to contamination of the river, canal, or other source, with acid effluents, to sulphuric acid from pyrites in pit water pumped out of coal pits, and even to organic acids, "peaty" in moorland waters percolated through masses of decomposing vegetable matter. Also may be due to the decomposition of animal or vegetable oil, added to the feed-water from condensed steam. The general remedy is to keep the feed-water alkaline and to use mineral oil.

3. *Acid-forming Salts*. As already indicated, magnesium salts are very troublesome in this respect, forming magnesium chloride, which decomposes into magnesium hydrate and free hydrochloric acid.

Aluminium salts also act in a similar manner, and care should be taken that no excess alumna ferric is used for grease removal. In the same way, ammonia salts are dangerous, such as ammonium chloride, or nitrate. The remedy again is to keep the feed-water alkaline.

4. *Electrolysis*. If, for example, a large amount of common salt (NaCl) is present in the feed-water, an electrolytic action is apt to be set up, the boiler acting as a huge electrolytic cell, and the water in the boiler containing salt acting as the liquid in the cell. The boiler plate then wastes away, causing "corrosion," which is really wasting. This action is apt to be very serious. An old remedy is to place in the bottom of the boilers, slabs or rolls of zinc connected to the boiler shell by heavy wires. This makes the electric current more complete, and a current flows from the shell of the boiler through the water in the boiler to the zinc plate and back through the connecting wires, and so on, completing the circuit. The zinc plate wastes away instead of the boiler plants, and is replaced as required.

Improved systems of preventing corrosion by sending a "reverse" electric current through the water are that of the "Cumberland," as supplied by Messrs. The Cumberland Engineering Co., Ltd., 6 Lloyds Avenue, London, E.C., and the "Gush" as supplied by Messrs. J. Downton & Co., Ltd., 69–71 West India Dock Road, Poplar, London, E.14.

OIL IN BOILER FEED-WATER.

Exhaust steam from engines blown into boiler feed-water contains oil from the lubrication of the engine, and the presence of this oil is most objectionable. In the first place, it is apt to form a thin film of oil on the boiler plant and prevent the water from coming properly in contact with the plate.

The plate, then, is in serious danger of overheating, causing bulging and collapse of furnace crowns. Insurance companies will not insure boilers using oil in the feed-water. According to Stromeyer, grease offers 1,000 times the resistance of steel to the passage of heat. Secondly, it floats on the top of the water in the boiler, and runs together into globules, which stick to the inside of the boiler at the water line. The heat of the plate above the water decomposes the oil, if animal or vegetable, into free organic acid, which "pits" the plates. With mineral oil, there is no action.

It is impossible to remove oil from water by mere mechanical filtration, because the oil is in such a finely divided condition. The only method is by entangling the oil in some precipitate, which can then be filtered easily.

The reagents in general use are alumna ferric and soda ash, which causes the oil in the water to become entangled in the precipitate of alumna hydrate; and on filtering through special filters of wood, wool quartz, coke, sand, etc., the oil is left behind.

An oil-eliminating plant works on the same general principle as a softening plant, using the alumna ferric and soda ash as the reagents instead of lime and soda. Such plants are supplied by all the makers of water-softening plants; also by Messrs. Bakers, Ltd., Hunslet, near Leeds.

In the *Davis-Perrett* Oil Eliminator, no chemicals at all are used, but an electrical process only. In this, the oily water passes between electrodes and the emulsive character of the oil is lost, so that it forms a flocculent precipitate easily filtered off. When finished, the water is said to contain less than 0·1 grs. of oil per gallon.

BOILER-FEED PUMPS

As typical of these, we will describe in some detail the "Weir" pump and the "Clarke-Chapman" pump.

THE "WEIR" BOILER-FEED PUMP.

This well-known single direct-acting boiler-feed pump is made in many various types for different purposes.

The steam valve consists of a main and an auxiliary valve. The main valve is for distributing steam to cylinders, and the auxiliary for distributing steam to work the main valve. The main valve works horizontally from side to side, being driven by steam admitted and exhausted from each end alternately. The auxiliary valve is actuated by lever gear from the rod of the pump, and moves on a face on the back of the main valve, and in a direction at right angles to the main valve. By this arrangement there is no dead centre.

These pumps work at very moderate speed, a very necessary factor in the life of a boiler-feed pump. Also by a special arrangement, they work very quietly because of a special type of steam valve; they slow down towards the end of the stroke and enable the water valves to settle quietly in their seats

STANDARD SIZES, CAPACITIES, ETC., OF WEIR DIRECT-ACTING FEED PUMPS

SIZE.	Capacity Gallons per hour.	Speed D.S. per min.	SIZE OF BRANCHES.				Floor Space.	Height.	Weght Packed for Shipment.
			Steam.	Exhaust.	Suction.	Discharge.			
Pump. Cyl. Stroke. In. In. In.			In.	In.	In.	In.	Ft. In. Ft. In.	Ft. In.	Lbs.
3½ × 5 × 6	515	22	¾	1	1½	1	1 7½ × 1 8	3 $7\frac{7}{16}$	568
4 × 6 × 8	810	20	¾	1	1¾	1½	1 9 × 1 10	4 3⅞	700
4½ × 6½ × 10	1,160	18	¾	1	2	1½	1 10 × 2 0½	4 10½	840
5 × 7 × 12	1,450	15	¾	1	2½	2	1 11 × 2 3	5 6¾	1,240
5½ × 8 × 12	1,730	15	1	1¼	2½	2	2 2 × 2 8	5 11	1,500
6 × 8½ × 12	1,950	14	1	1¼	3	2½	2 3 × 3 0	5 10½	1,708
6 × 8½ × 15	2 430	14	1	1¼	3	2½	2 3 × 3 1	6 7½	1 820
6 × 8½ × 18	2,700	13	1¼	1½	3	2½	2 5 × 3 1	7 4½	2,124
7 × 9½ × 15	3,320	14	1¼	1½	3½	3	2 7½ × 3 5½	7 1¾	2,548
7 × 9½ × 18	3,700	13	1¼	1½	3½	3	2 7½ × 3 5½	7 10¾	2 772
7 × 9½ × 21	4,000	12	1½	2	3½	3	2 8½ × 3 5½	8 $7\frac{11}{16}$	2,912
8 × 10½ × 18	4,500	12	1½	2	4	3½	2 10 × 3 6½	7 11⅝	2,968
8 × 10½ × 21	5,250	12	1½	2	4	3½	2 10 × 3 6½	8 8⅝	3,164
8 × 10½ × 24	6,000	12	1½	2	4	3½	2 10 × 3 6½	9 5⅝	3,416
9 × 12 × 21	6,630	12	1½	2	4½	3½	2 11½ × 3 9	9 0½	3,787
9 × 12 × 24	7,540	12	1½	2	4½	3½	2 11½ × 3 9	9 9½	3,948
9½ × 12½ × 24	8,400	12	1¾	2	5	4	2 11 × 4 0	10 0	4,788
10 × 13½ × 24	9,300	12	1¾	2¼	5	4	2 11½ × 4 0	10 1½	4,984
10 × 13½ × 26	10,000	12	1¾	2¼	5	4	3 0 × 4 6	10 2	5,376

No.	
1.	Steam slide valve chest
2.	Double joint
3.	Front stay
4.	Bottom spindle
5.	Valve gear levers
6.	Front stay bush
7.	Ball crosshead
8.	Main crosshead
9.	Crosshead pin
10.	Piston rod
11.	Piston body
12.	Piston rings
13.	Cylinder cover
14.	Discharge valve seat
15.	Discharge valve seat ring
16.	Suction valve seat
17.	Suction valve guard
18.	Discharge valve guard
19.	Water valves
20.	Bucket
20A.	Bucket rings
21.	Pump rod
22.	Pump cover
23.	Valve chest cover
24.	Steam stop valve
26.	Auxiliary valve spindle
27.	Cylinder neck ring
28.	Cylinder gland.
29.	Pump neck ring
30.	Pump gland

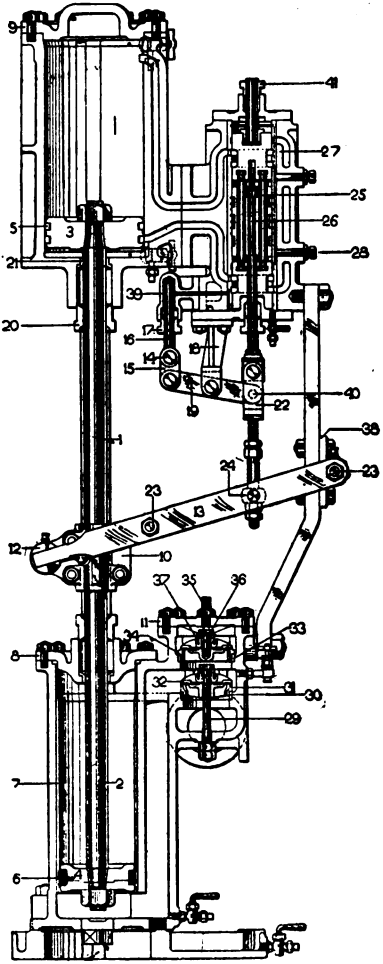
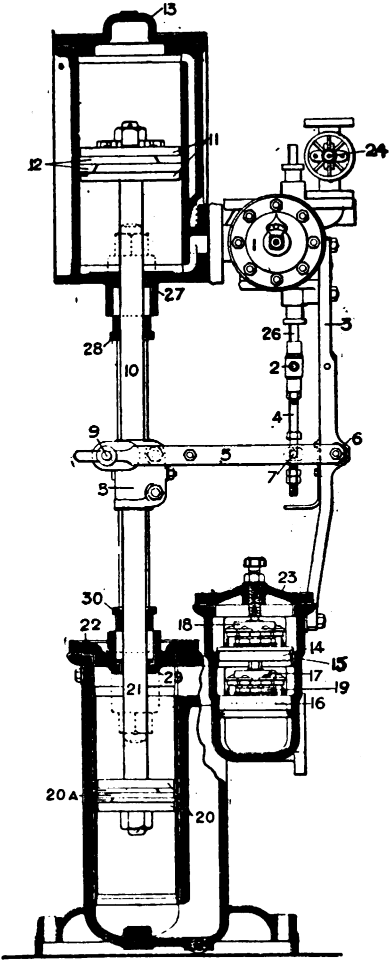


Fig. 30.—The Weir Standard Feed Pump.
For Land Installations—Sectional View.



1. Piston rod, with nut
2. Plunger rod, with nut
3. Piston body
4. Plunger body
5. Piston ring
6. Plunger ring
7. Water cylinder liner
8. Water cylinder cover
9. Steam cylinder cover
10. Button, complete with bolts
11. Water valve box cover
12. Button sleeves
13. Side lever
14. Gear bolts
15. Link plates
16 Balance piston
17. Bottom spindle glands
18. Fulcrum.
19. Rocking lever
20. Piston rod or plunger rod glands
21. Piston rod or plunger rod neck bushes
22. Tappet rod complete with nuts and gear bolt
23. Side-lever spindles
24. Striking piece
25. Main valve columns
26. Main slide spindle, complete
27. Steam chest, complete
28. Cushioning bolt and nut
29. Suction valve bolt and nut
30. Suction valve ring
31. Suction valve seat
32. Suction or discharge valve lids
33. Discharge valve ring
34. Discharge valve seat
35. Top bolt and nut
36. Discharge valve spindle
37. Water valve spring
38. Gear bracket
39. Balance cylinder
40. Plain gear pin
41. Tail rod gland

Fig. 31.—Clarke-Chapman Feed Pump (Woodeson Patent).

without shock. The steam cylinder is also worked on the expansive principle, so that the steam valve cuts off before the end of the stroke, so that the pump is economical for steam consumption. Also there is no dead centre and the pump starts at any part of the stroke.

THE "CLARKE-CHAPMAN" PUMP (*Woodeson patent*).

The speciality of this pump is the Woodeson patent steam-through-piston valve (NP type), Fig. 31, for detailed description of which the firm's catalogues must be consulted.

TURBO PUMPS

For large installations, centrifugal boiler-feed pumps are now being more extensively used, especially steam turbine-driven pumps. They have the advantage of taking up very little room; they are noiseless; the feed is absolutely steady; when the boilers are shut down, no dangerous rise in pressure takes place; no safety valves are required; they are automatic in action; and all the exhaust steam can be passed back direct to the boiler-feed water, as it contains no oil, so that they are extremely economical in working.

Such pumps Fig. 33 consist of a steam turbine of the impulse type, having one pressure and several velocity stages direct coupled to a single-stage centrifugal pump of special design. The impeller of the pump is carried on the overhanging end of the turbine shaft, so that no third bearing is necessary. The turbine casing and base of the combined set is cast iron and the pump is of bronze. The turbine glands are provided with carbon packing, and the turbine bearings are of the self-oiling ring type. The pump is fitted with an emergency governor directly attached to the turbine shaft, so that if the water supply failed and the pump speed increases, the whole set is shut down by the governor throttling the steam inlet. In practice, the pump adjusts itself automatically to changes in both steam and water. A big advantage is that the exhaust steam from the turbine contains no oil, so that it can be passed direct into the feed-water supply tank. In consequence, the actual power required to drive the pump is practically nil.

INJECTORS

Injectors are generally used for single boilers and as a stand-by in case of breakdown of the boiler-feed pump. They are not able to take water over about 120° F.

The theory of the working of the injector is somewhat complicated, but it consists essentially of a jet of stem at high velocity surrounded by cool water. The steam is condensed in the cone of the injector, which causes a vacuum, and the momentum attained is sufficient to force a stream of hot water into the boiler against the boiler pressure. Practically the whole of the heat is returned to the boiler, except the small amount of energy used.

Injectors are apt to get out of order very easily. This is often due to leaky pipe connections and entry of air, which prevents the formation of a vacuum. Also the steam may be

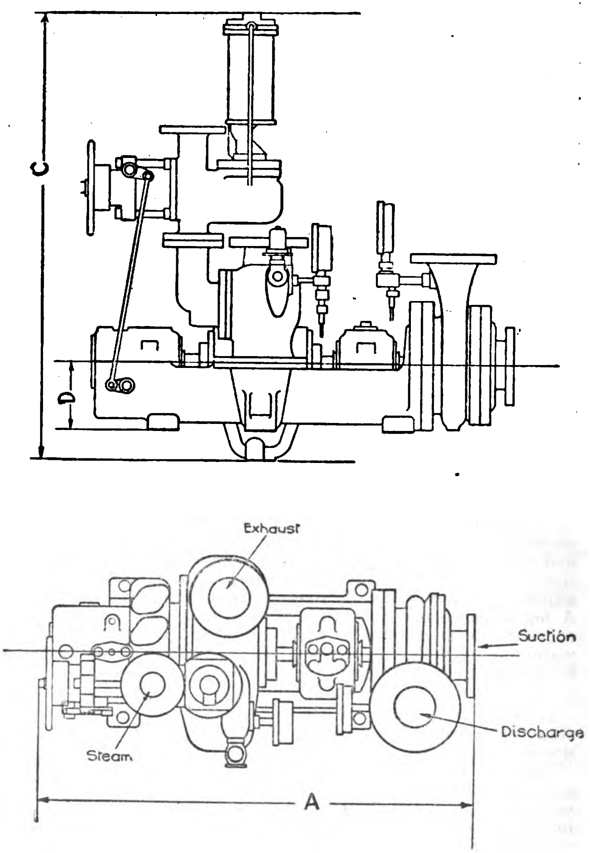


FIG. 32.—WEIR TURBO PUMP.

TURBO PUMP DATA

TYPE.	Capacity.	Steam.	Exhaust.	Suction.	Discharge.	DIMENSIONS (*see* Illustration).											Weight.
						A.		B.		C.		D.	E.		G.		
	Galls. per hr.	Ins.	Ins.	Ins.	Ins.	Ft.	Ins.	Ft.	Ins.	Ft.	Ins.	Ins.	Ft.	Ins.	Ft.	Ins.	Lbs.
TFP 7..	7,000	1½	3	3	3	3	8½	2	0	3	7¾	6¼	3	5	2	1¼	870
TFP 10..	10,000	1½	3	3½	3½	3	8¾	2	0	3	7¾	6¼	3	5	2	2¼	910
TFP 15..	15,000	2	4	4	4	3	10¾	1	11¼	3	11¾	7¼	3	8¾	2	4½	1,120
TFP 20..	20,000	2	4	6	5	3	11	1	11¼	3	11¾	7¼	3	8¾	2	9	1,170
TFP 27..	27,000	3	5	6	6	4	5½	2	2¼	4	7¾	9	4	4¼	3	1½	1,840
TFP 35..	35,000	3	5	7	7	4	6½	2	2¼	4	7¾	9	4	4¼	3	6½	1,960

very wet or not sufficient, and it is best to work the injector with a separate steam pipe direct from the boiler. Further, grit and dirt is easily deposited on the cones.

There are various types of injectors on the market. The "self-acting" (automatic) type works automatically when the steam and water are turned on. In the "non-automatic" type, the steam and water have to be regulated separately by hand. Many injectors are made with adjustable cones to suit varying conditions, and "exhaust steam" injectors are worked with exhaust steam supplemented by live steam when necessary.

FEED-WATER REGULATORS

The advantages of feed-water regulators are that, in a given boiler, there is a certain level of water at which the fullest efficiency is obtained. If the level of the water falls, the boiler does not absorb the full amount of heat; and if too much water is added, the rate of evaporation is reduced, the pressure falls, and the fires have to be forced, with loss of efficiency. Hand attention does not, as a rule, maintain a constant level.

It is claimed that an efficient automatic regulator will aid the safety of the boiler from low water, prevent wet steam due to too high a level, and reduces unequal expansion and contraction. In the case of water-tube boilers, the use of an automatic regulator is almost essential, because of the small amount of water in the boiler, and the danger of a sudden demand for steam seriously reducing the level of the water before the attendant can get to the feed-inlet regulating valve.

Whatever type is selected, one point essential is that in case of accident, either the regulator must not stop the inlet of the water, or a by-pass must be installed so that hand regulation can be resorted to at once. Also in the case of most types of feed pumps, it is much better to install the regulator along with the boiler-feed pump regulator in addition, so that the speed of the pump is directly controlled by the feed-water regulator.

The "Anthony" Regulator consists of a float in a chamber attached to the boiler, so that the chamber includes the water level in the boiler. As the water level rises and falls, the water in the chamber does the same, the steam space above also varying correspondingly. The float actuates a small steam valve, which supplies live steam to a special valve placed in the water inlet pipe to the boiler. As the float rises or falls, it opens less or more steam to the special valve, which closes or opens the feed-water inlet pipe to the boiler, thus regulating the amount of water admitted to the boiler, depending on the water level.

The "Babcock and Wilcox" Regulator consists of a hollow cylinder, which is rotated on its axis by means of a float and lever inside the boiler itself. The cylinder has ports cut diagonally along its walls, and works in a liner having corresponding ports. The liner is fitted in a casing attached to the boiler shell. The feed-water enters the boiler through this apparatus and, as the float in the boiler rises or falls, the motion is communicated through the boiler shell by means of a rod, and moves the liner, which admits more or less water to the boiler.

The "Cope" Regulator depends on the expansion and contraction of a heavy metallic tube mounted on a base, and connected by a lever and strut to a control valve in the feed line. The expansion tube is set in an inclined position opposite the water level in the boiler, and is connected to the steam and water spaces. The water level moves up and down in the inclined tube as n the boiler, and the upper portion of the tube contains live steam and the lower portion water cooled by radiation. This inclined tube expands and contracts, depending on the amount of the hot steam contained in it, that is in proportion to the water level, and this expansion and contraction works a control valve on the feed-water inlet pipe to the boiler.

The "Crosby" Regulator consists of a special apparatus called a "power producer," which is attached at one portion to the steam section of the boiler drum and at a lower portion to the water section, the water level in the boiler corresponding to the level in the "power producer." The "power producer" contains also a closed vessel containing water, and the live steam or water, according to the level in the boiler, heats this closed vessel and generates a small amount of steam. This effects another water column, which actuates directly a special control valve on the feed-water inlet pipe. This valve, therefore, is controlled actually by the water level in the boiler.

The "Thermofeed" Regulator consists of a float chamber attached to the boiler by a pipe at the bottom connected to the water space, and by a pipe above connected to the steam space, so that the water level in the float chamber corresponds to the boiler level. In this chamber is a float which regulates a small live steam pipe connected to a control valve on the feed-water inlet pipe. As the water in the boiler rises and falls, the float admits more or less live steam to the control valve, which rises or falls and varies the amount of water admitted.

FEED-WATER HEATERS

A valuable aid to boiler economy is the utilization of any exhaust steam to heat the feed-water before entering the boiler, or economizer. Feed-water heaters are constructed on the general principle that the exhaust steam passes through a nest of tubes, whilst the water flows over the outside of the tubes in a vertical mild steel casing. The latter is fitted with suitable cleaning arrangements, with safety-valve and blow off cock.

ECONOMIZERS

The fuel economizer for steam boilers was introduced about seventy years ago, being invented by Mr. Edward Green in 1845, when the first patent was taken out. It consists essentially of a series of vertical cast-iron tubes placed in the hot exit gases from the boilers, and so arranged that the boiler-feed water passes through the tubes on its way to the boilers, thus absorbing a large proportion of the heat in the gases which would otherwise be wasted.

The tubes are provided with automatic scrapers to keep the outside surface free from soot and dirt.

The economizer pipe is usually of a standard dimension of 9 ft. length and sometimes 12 ft., and $4\frac{3}{16}$ ins. external diameter, being constructed to suit the boiler pressure and tested by hydraulic pressure to, say, 200 lbs. over working pressure. Each of these standard pipes has approximately 10 sq. ft. of heating surface, holds $6\frac{1}{4}$ galls. of water, and weighs $\frac{1}{8}$ ton. The power required to drive the scrapers is very small, being, for example, about $\frac{1}{4}$ h.-p. for a set of ninety-six tubes. Cast-iron is the most suitable material for the tubes, at any rate up to 200 lbs. steam pressure, as steel, wrought iron, copper, brass, etc., will not resist so well the corrosive action of the hot gases.

As regards the number of tubes to be installed for the best results on any given boiler-plant, the makers usually recommend an installation large enough to hold one hour's supply of feed-water in the pipes. Thus if a boiler is evaporating 650 galls. per hour, each tube holds $6\frac{1}{4}$ galls., and the number of economizer tubes would be then 104. Because of the gearing arrangement, economizers can only be made in certain standard sections of 6, 8, or 10 tubes wide, the wider sections being used for larger boiler plants. A plant of one boiler only would have a standard 6-tube wide economizer, and the nearest standard size to 104 tubes would be 96 tubes, being 6 tubes wide in two sections of 8 tubes in length. The next size or grouping would be 120 tubes, in twenty sections 6 tubes wider Larger sets of economizers would be 8 tubes wide or 10 tubes wide to diminish the resistance to the draught. It is customary to make a rough calculation on the basis of 4 pipes for every ton of coal burnt per week of 54 hrs., or 1 pipe for every 3 i.h.p. developed, 4 sq. ft. of heating surface per boiler h.p., and so on. The advantages of the economizer are as follows—

1. The Saving in the Coal Bill.

By utilizing some of the waste heat of the flue gases in heating up the boiler feed-water.

The results of 250 boiler plant tests by the Author are given in the following table, dividing the plants into six divisions, according to the saving obtained.

In actual practice, therefore, only about 17 per cent. of the plants fitted with economizers were saving 15 per cent. or over of the coal bill. Over 30 per cent. were saving less than 10 per cent.

The reason for such an average figure as 11·4 per cent. saving is, of course, not due to the fault of the economizer as an

RESULTS OF WORKING WITH ECONOMIZERS

Division according to Saving obtained by the economizer.	No. of Plants.	Average Temperature of Feed Water.		Average Temperature of Flue Gases.		Draught in Inches W.G.		Steam Pressure Gauge.	Total Evaporation on Plant per hour.	No. of Tubes on Plant.
		Before.	After.	Before.	After.	Chimney Base.	Side Flue or Down-take of Boilers.	Lbs. per sq. inch.		
No economizer at all	95	° F. —	° F. —	° F. —	° F. —	—	—	—	Lbs. —	—
Less than 5 % ..	14	134	172	568	371	0·85	0·41	80·0	17,616·0	228
5 % to 7½% ..	15	123	194	576	423	0·61	0·35	79·0	15,340·0	203
7½% to 10 % ..	23	119	261	549	374	0·74	0·33	100·0	16,630·0	231
10 % to 12½% ..	33	118	243	583	403	0·88	0·39	106·0	29,723·0	401
12½% to 15 % ..	46	110	262	584	379	0·99	0·41	99·0	19,760·0	280
Over 15%	24	170	284	610	375	1·35	0·60	119·0	23,497·0	392
Total Plants ..	250									

DETAILED RESULTS OF WORKING OF ECONOMIZERS IN 24 PLANTS SAVING OVER 15 PER CENT.

No. of Plant.	Average Temp. of Feed Water.		Average Temp. of Flue Gases.		Draught in Inches.		Steam Pressure Gauge.	Total Evaporation on Plant per Hour.	Number of Tubes in Economizer on Plant.	Per Cent. Saving due to Economizers.
	Before Economizers.	After Economizers.	Before Economizers.	After Economizers.	W.G. Chimney Base.	Side Flue or Downtake Boilers.				
	° F.	° F.	° F.	° F.			lb. per sq. in.	lb.		per cent.
1	100	321	612	344	1·40	0·50	180·0	18843·0	480	19·6
2	59	282	548	328	1·52	0·98	81·0	41166·0	560	19·3
3	95	311	680	375	3·10	0·80	161·0	37785·0	488	19·2
4	65	281	605	475	1·50	0·80	76·0	18174·0	288	18·8
5	88	289	662	372	0·67	0·28	97·0	12882·0	394	17·8
6	60	266	625	420	0·45	0·85	79·0	16395·0	216	17·8
7	112	309	565	393	0·45	0·75	98·0	5310·0	120	17·8
8	90	301	609	366	1·70	0·82	139·0	72516·0	960	17·6
9	115	303	585	324	1·00	0·60	119·0	14394·0	192	17·4
10	110	308	689	448	1·20	0·86	104·0	11578·0	320	17·4
11	104	296	616	377	0·60	0·40	74·0	6032·0	144	17·3
12	92	289	612	336	1·70	0·66	189·0	16118·0	288	17·3
13	151	333	463	370	0·61	0·42	160·0	11830·0	320	16·9
14	91	275	609	377	1·00	0·59	95·0	7968·0	192	16·3
15	105	287	759	525	1·20	0·60	103·0	29484·0	640	16·3
16	138	307	715	353	1·75	0·45	120·0	41435·0	640	16·0
17	79	259	609	450	0·92	0·55	111·0	12908·0	320	15·8
18	129	302	487	325	1·68	0·52	154·0	6390·0	160	15·7
19	114	286	603	387	1·50	0·60	112·0	24808·0	384	15·6
20	109	280	749	550	1·50	0·70	103·0	8734·0	96	15·4
21	107	276	597	328	1·15	0·30	81·0	25264·0	409	15·3
22	115	281	539	463	0·90	0·30	72·0	7676·0	168	15·2
23	117	284	588	394	1·15	0·55	88·0	18219·0	280	15·2
24	88	260	599	821	1·24	0·35	157·0	98032·0	1,408	15·1

GREEN'S PATENT FUEL ECONOMIZER

PERCENTAGE OF SAVING EFFECTED BY HEATING FEED WATER

Final Temperature.	Initial Temperature of Water.											
	40°	50°	60°	70°	80°	90°	100°	120°	140°	160°	180°	200°
100	5·1	4·285	3·456	2·616	1·758	·887	·00					
120	6·8	5·999	5·184	4·36	3·516	2·661	1·79	·00				
140	8·5	7·713	6·912	6·104	5·274	4·435	3·58	1·822	·00			
160	10·2	9·427	8·64	7·848	7·032	6·209	5·37	3·644	1·858	·00		
180	11·9	11·141	10·368	9·592	8·79	7·983	7·16	5·466	3·716	1·892	·00	
200	13·6	12·855	12·096	11·336	10·548	9·757	8·95	7·288	5·574	3·784	1·93	·00
220	15·3	14·569	13·824	13·08	12·306	11·531	10·74	9·11	7·432	5·676	3·86	1·968
240	17·0	16·283	15·552	14·824	14·064	13·305	12·53	10·932	9·29	7·568	5·79	3·936
260	18·7	17·997	17·28	16·568	15·822	15·079	14·32	12·754	11·148	9·46	7·72	5·904
280	20·4	19·711	19·008	18·312	17·58	16·853	16·11	14·576	13·006	11·35	9·65	7·872
300	22·1	21·425	20·736	20·056	19·338	18·627	17·9	16·398	14·864	13·24	11·58	9·84

ECONOMIZERS. PERCENTAGE OF SAVINGS

The following Table gives the percentage of saving for each degree of increase in temperature of feed-water heated. Pressure of steam in boiler, lbs. per sq. inch above atmosphere.

Initial Temp. of Feed.	Working Pressure in Boiler. Pounds per Square Inch.											Initial Temp. of Feed.
	0	**20**	**40**	**60**	**80**	**100**	**120**	**140**	**160**	**180**	**200**	
32°	·0872	·0861	·0855	·0851	·0847	·0844	·0841	·0839	·0837	·0835	·0833	32°
40	·0878	·0867	·0861	·0856	·0853	·0850	·0847	·0845	·0843	·0841	·0839	40
50	·0886	·0875	·0868	·0864	·0860	·0857	·0854	·0852	·0850	·0848	·0846	50
60	·0894	·0883	·0876	·0872	·0867	·0864	·0862	·0859	·0856	·0855	·0853	60
70	·0902	·0890	·0884	·0879	·0875	·0872	·0869	·0867	·0864	·0862	·0860	70
80	·0910	·0898	·0891	·0887	·0883	·0879	·0877	·0874	·0872	·7870	·0868	80
90	·0919	·0907	·0900	·0895	·0888	·0887	·0884	·0883	·0879	·0877	·0875	90
100	·0927	·0915	·0908	·0903	·0899	·0895	·0892	·0890	·0887	·0885	·0883	100
101	·0936	·0923	·0916	·0911	·0907	·0903	·0900	·0898	·0895	·0893	·0891	110
120	·0945	·0932	·0925	·0919	·0915	·0911	·0908	·0906	·0903	·0901	·0899	120
130	·0954	·0941	·0934	·0928	·0924	·0920	·0917	·0914	·0912	·0909	·0907	130
140	·0963	·0950	·0943	·0937	·0932	·0929	·0925	·0923	·0920	·0918	·0916	140
150	·0973	·0959	·0951	·0946	·0941	·0937	·0934	·0931	·0929	·0926	·0924	150
160	·0982	·0968	·0961	·0955	·0950	·0946	·0943	·0940	·0937	·0935	·0933	160
170	·0992	·0978	·0970	·0964	·0959	·0955	·0952	·0949	·0946	·0944	·0941	170
180	·1002	·0988	·0981	·0973	·0969	·0965	·0961	·0958	·0955	·0953	·0951	180
190	·1012	·0998	·0989	·0983	·0978	·0974	·0971	·0968	·0964	·0962	·0960	190
200	·1022	·1008	·0999	·0993	·0988	·0984	·0980	·0977	·0974	·0972	·0969	200

appliance, but is due to the fact that the economical generation of steam on scientific lines is not understood, and economizers generally are not installed on correct lines. The economizer is, of course, an absolutely essential part of any up-to-date boiler plant, and the saving obtained should be 17½ per cent. to 20 per cent. under average conditions. In this connection, the figures of each of the separate twenty-four plants found to be saving 15 per cent. or over, given in Table II, will be of interest showing what is actually being done on many boiler plants

The great increase in temperature of the feed water and the reduction in temperature of the flue gases will be noted.

The average figures for the whole 155 typical plants were feed-water of 115° F. raised to a temperature of 241° F., and flue gases reduced from 581° F. to 389° F.

The methods of calculating the saving have already been described in detail but the table on page 714, is very handy for quick reference, and explains itself.

Also for very rough calculations, 11°F. in the feed-water can be said to be equal to 1 per cent. saving in the coal bill.

2. Reduction in the strain on the boilers by the provision of a high temperature feed-water preventing unequal expansion and contraction.

3. Increases the evaporative capacity of the boilers, since the water enters the boiler at a higher temperature.

DISADVANTAGES.

The economizer, being a machine, has some disadvantages, as follows—

1. *A serious choking of the draught* due to the cooling of the flue gases. Taking natural draught first, the draught in a chimney, other things being equal, is simply dependent on the mean temperature of the gases in the chimney. Thus, for example, taking a brick chimney 170 ft. high with a temperature of 600° F., the temperature of the gases in the chimney is reduced about 2° F. for every 3 ft. in height by the cooling action of the air, so that the mean temperature of the hot gases in the chimney will be about 544° F., taking the temperature of the air as 60° F. The actual draught in inches water-gauge at the chimney-base is given very nearly by the following formula (which allows for friction losses)—

$$P = H\left(\frac{7\cdot6}{t} - \frac{7\cdot9}{T}\right)$$

where P = inches water-gauge;
H = height of chimney in feet above firing level;
T = mean absolute temperature of chimney gases;
t = absolute temperature of atmosphere;

$$P = 170 \times \left(\frac{7\cdot6}{60 + 461} - \frac{7\cdot9}{544 + 461}\right)$$

$$= 170 \times \left(\frac{7{\cdot}6}{521} - \frac{7{\cdot}9}{1{,}005}\right)$$
$$= 170 \times (0{\cdot}01458 - 0{\cdot}00786)$$
$$= 170 \times (0{\cdot}00672)$$

= 1·14 ins. W.G. corresponding to, say, 0·60 in. W.G. in the side flues of a "Lancashire" boiler.

If, now, we install economizers and reduce the coal bill, say, 17½ per cent., the temperature of the gases will be reduced from 600° F. to, say, 350° F.

The mean temperature in the chimney will now be only about 294° F., and the draught will be—

$$P = \left(\frac{7{\cdot}6}{t} - \frac{7{\cdot}9}{T}\right)$$
$$= 170\left(\frac{7{\cdot}6}{60 + 461} - \frac{7{\cdot}9}{294 + 461}\right)$$
$$= 170\left(\frac{7{\cdot}6}{521} - \frac{7{\cdot}9}{755}\right)$$
$$= 170\,(0{\cdot}01458 - 0{\cdot}01046)$$

= 0·70 ins. W.G. corresponding to, say, 0·35 ins. W.G. in the side flues of a "Lancashire" boiler.

Thus, again taking the average figures for the 155 typical plants, the approximate draught in the chimney-base was 1·10 ins. W.G., and the economizers in saving 11·4 per cent. of the coal bill, reduced this draught to 0·60 ins. W.G. in the downtake of the boiler. There is obviously, therefore, a very serious drop in the draught, which causes certain difficulties.

In some cases, it is necessary to open slightly the by-pass dampers and allow a portion of the hot gases to go straight up the chimney to waste, so as to maintain a draught to burn the amount of coal required to keep up steam.

2. *Scale Formation.* The formation on the inside of the tubes of a hard deposit of scale from impurities in the feed-water, which prevents the conduction of the heat from the hot gases, and is very liable to cause the tubes to be burnt out, and the obvious remedy is to soften the feed-water.

3. *Corrosion.* A further source of trouble of the economizer is corrosion of the pipes. This is caused by the condensation of the acid flue gases on the outside of the tubes at the lower end where the cold water enters. Flue gases contain sulphurous acid from the sulphur in the coal, which condenses because of the great difference in temperature between, say, 600° F. of the gases and 50° F. of the inlet water. This corrosive action is intensified by excessive sulphur in the coal and moisture in the flue gases due to damp flues or by excessive amount of steam being used for steam-jet furnaces. One remedy is to ensure

that the feed-water does not enter the economizer at a temperature of less than 100° F., by providing an automatic circulating arrangement to take continuously a small portion of the water from the hot end and mix it with water from the cold end.

4. *Explosion Dangers.* The safety valves may fail. There is the danger due to corrosion or wasting away, so that the tubes wear thin and then collapse. Also explosions have been caused by the economizer being shut out of the feed circuit and the flue gases still partially circulating round the tubes due to defective dampers. As a consequence, the empty tubes become red hot unknown to the attendants, and when cold feed-water is pumped in an explosion results.

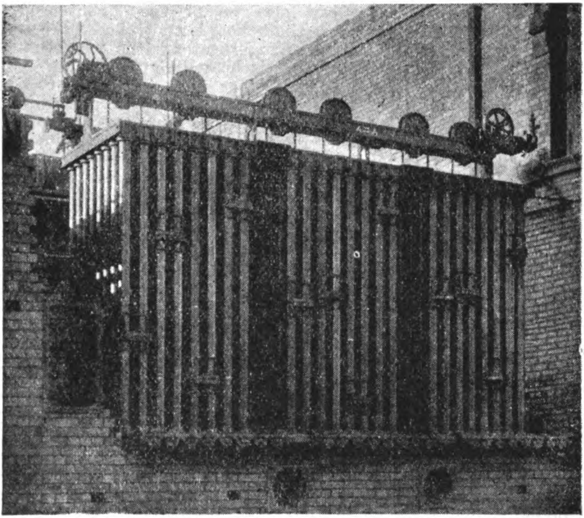


FIG. 33.—CLAY CROSS ECONOMIZER, LITCHURCH GASWORKS, DERBY.

In any case, the economizer should be regularly inspected like the boiler.

Economizers can be applied not only to boilers of every description, but to any other sources of waste heat, such as refuse destructors, blast furnaces, coke ovens, cement kilns, etc., etc.

Also the economizer tubes can be arranged in addition to boiler-feed water heating, to provide an ordinary non-pressure hot water supply for general factory processes, or for air heating ; and many economizer installations are working on these lines.

SUPERHEATERS

The saturated steam from the boiler is passed through a series of seamless steel tubes, which are heated by the hot gases from the boiler exit before passing to the economizers. Thus in "Lancashire" or similar boilers, the superheater is placed in the downtake, and in water-tube boilers between the tubes in some convenient position depending on the type of boiler. The valves need to be cast-iron or nickel alloy, and it is not now the usual practice to erect superheaters with a by-pass. Superheaters can also be of the "controlled" type, in which the heat from the boilers is controlled by means of dampers. A further type of superheater is the "externally fired," in which the superheater is heated by a separate coal or coke furnace and not by the boiler gases.

A superheater must be designed so as to prevent burning out by a chance overheating; to ensure a proper circulation of the steam; to be easily repaired, and spare parts quickly replaced; the various parts, external and internal, to be accessible, and to be capable of free expansion and contraction. The number of tubes and heating surface in any particular superheater depends on the amount of superheat required, the amount of steam passing, and the quality of the coal. Very roughly, the heating surface of a superheater is about 20 per cent. of the heating surface of the boiler, and the weight of a superheater for a standard 30′ × 8′ "Lancashire" boiler is about 2 tons.

Superheating is applied in two general ways, namely, high superheating for engine or turbine efficiency, and partial superheat to dry the steam. Theoretically, steam can, of course, be superheated to about 1,000° F. when it decomposes. For modern boiler and power plant, a common figure is 200° F. of superheat, which at 200 lbs. gauge pressure would equal, say, 590° F. in the steam, and there is a tendency to go even higher, up to 650° F.

It is claimed by superheater makers generally that superheating will save 10–25 per cent. of the coal bill on the average plant. Roughly speaking, superheating 100° F. will save 10 per cent. in turbines, 12 per cent. in triple engines, and 18 per cent. in simple engines. The greatest saving is in the engine cylinder, the average cylinder condensation being 20–40 per cent. of the steam used, which is considerably reduced by superheating.

Partial superheating to 75–100° F. is also very valuable in thoroughly drying the steam and preventing condensation losses in long pipe circuits.

In superheating, steam, of course, expands considerably; for example see page 719.

This fact is often overlooked in applying superheated steam to an engine running on saturated steam.

In the "Unit" Superheater (Superheater Units, Ltd., 4 Castle Square, Swansea), as applied to a Lancashire boiler, the saturated steam enters the front portion A, Fig. 34, thence up

Pressure in lbs. per sq. in.	Temperature on the Fahr. Scale in degrees.	Volume of 1 lb. of Saturated Steam in cub. ft.	Volume of 1 lb. of Steam superheated 100° F. in cub. ft.	Volume of 1 lb. of Steam superheated 150° F. in cub. ft.	Volume of 1 lb. of Steam superheated 200° F. in cub ft.
100	338	3·81	4·38	4·64	4·87
120	350	3·27	3·76	3·98	4·18
160	370	2·91	2·94	3·12	3·27
180	379	2·31	2·63	2·81	2·95
200	386	2·10	2·41	2·56	2·68

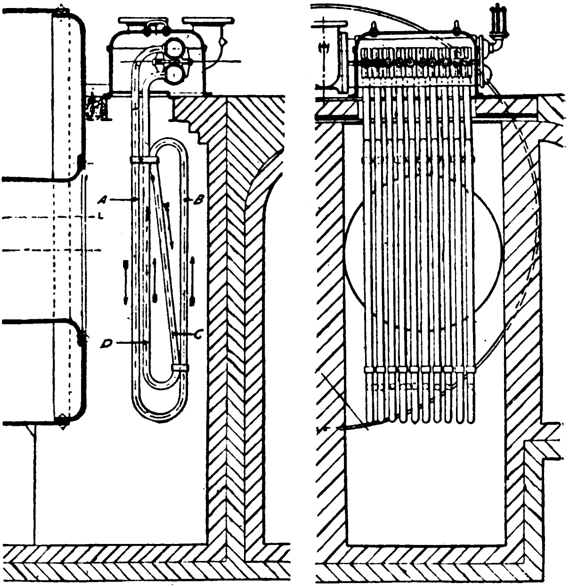


FIG. 34.—UNIT SUPERHEATER ON LANCASHIRE BOILER.

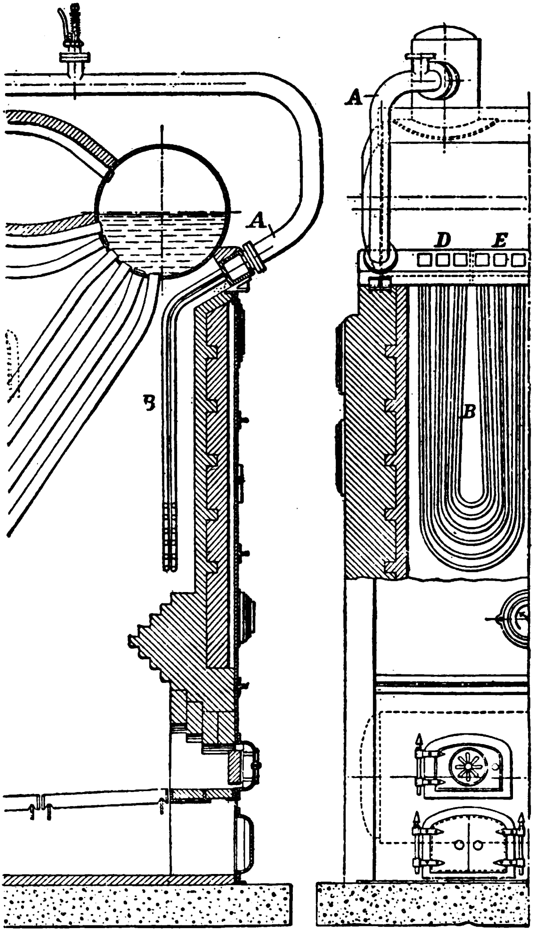


FIG. 35.—SUGDEN SUPERHEATER ON STIRLING BOILER.

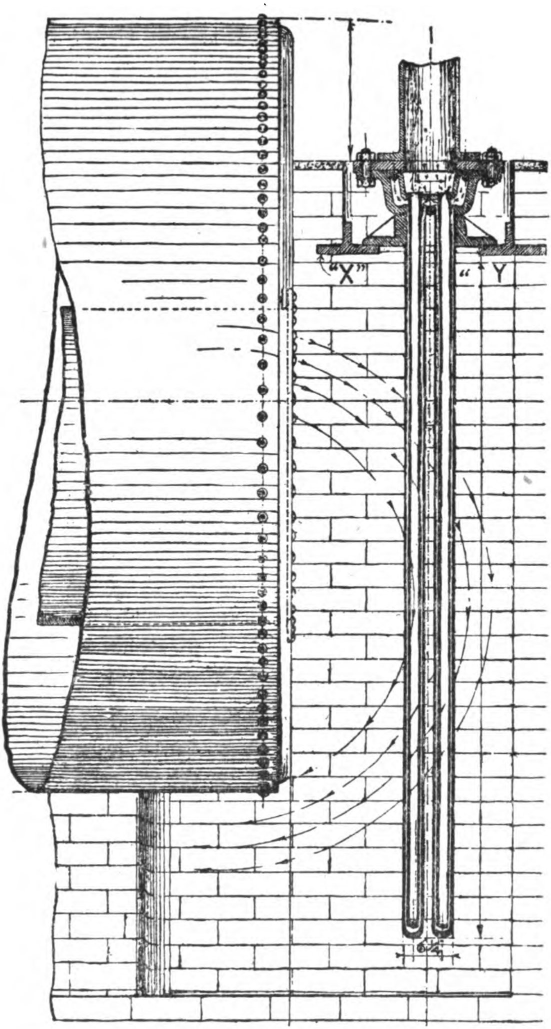


FIG. 36.—BOLTON FIELD TUBE SUPERHEATER.

the portion B, down the leg C, being gradually superheated in its passage, and lastly passes up the section D. The tube joints are out of the path of gases, and comprise steel collars held up by dogs which facilitate assembly and removal.

The "Sugden" Superheater (T. Sugden, Ltd., 180 Fleet Street, London), as applied to a Stirling boiler, consists in two double groups of U-tubes B, Fig. 35 (*see* p. 720), expanded into a header with two compartments, D E, and placed in the first chamber, where it is easy of access for cleaning or removal.

In the "Bolton" Superheater (Bolton's Superheater, Ltd., Heatizer Works, Trafford Park, Manchester), Field tubes, Fig. 36 (*see* p. 721) are used. The headers rest on two cross beams XY, and are suspended as shown at the rear of the boiler. Other superheaters are shown on pp. 631, 638, 645, 648, and 651.

TESTING OF BOILER PLANTS

It is absolutely essential, in order to obtain the best results on any boiler plant, to know exactly what the plant is doing.

It requires to be clearly understood that the main object of a test is to find out the figures for the performance of the plant as usually worked from week to week. Many tests have been carried out under purely abnormal conditions, such as by specially repairing leaky brickwork, supplying better quality fuel, reducing the steam demand, and so on. Boiler tests are also of great value in special cases, as, for example, in determining the exact value under practical conditions of different qualities of fuel, the efficiency when on overload, efficiency when on reduced load, and so on.

1. Description of the Method of Carrying Out a Boiler Test.

The Author recommends from many years' continuous experience the following method of carrying out a boiler test.

First, a fully-detailed test of one working day or shift, say, 8–12 hrs., or 24 hrs. if the plant works day and night, followed by a long check test of at least one week. A test of a few hours only is of little use. The data can be summarized—

A. *Day's Test.* General description of the boiler plant—

A. Type of boilers.
B. Number of boilers worked.
C. Dimensions of boilers. (If tubular boilers, state heating surface and rated evaporation per hour.)
D. Grate area (excluding dead plate).
E. Method of firing. (If mechanical, state make of stoker and h.p. taken to drive same.)
F. Type of fire-bar (short description).
G. If steam jets used, short description.
H. Economizer plant :
    No. of tubes. Height of tubes.
    No. of tubes wide. Arrangement.
I. Chimney :
    Height.
    Inside dimensions, top. Inside dimensions, bottom.

J. Method of draught. (If mechanical, give short description, stating average h.p. used.)
K. Superheaters, short description.
L. Boiler-feed pump, type.
M. Injector.
N. Source of feed-water.
O. Pressure at which boilers blow off.
P. Lowest working pressure permissible for proper working of factory.
Q. Number and wages of firemen per shift.
R. Number and wages of ash wheelers per shift.
S. Number of shifts per week.
T. Remarks.

2. PARTICULARS RELATING TO BURNING OF FUEL.

A. Quality of fuel used.
B. Price of fuel used (in firehole).
C. Analysis of fuel used :
B.Th.U. as fired
Ash. Water.
D. Amount of fuel used.
E. Fuel burned per boiler per hour.
F. Fuel burned per square foot of grate area per hour.
G. Draught (chimney base).
H. Draught (back flues of boilers).
I. Density of black smoke.
J. Temperature of flue gases before economizers.
K. Temperature of flue gases after economizers.
L. Per cent. CO_2 in flue gases (continuous record).
M. Analysis of gases, CO_2, CO. O. N.

3. PARTICULARS RELATING TO STEAM PRODUCTION.

A. Total water evaporated. lbs.
B. Variation in demand for steam per hour.
C. Water evaporated per boiler per hour, gallons.
D. Water evaporated per sq. ft. grate area per hour.
E. Method of measuring water.
F. Average temperature of feed-water before economizers.
G. Average temperature of feed-water after economizers.
H. Percentage saving due to economizers.
I. Analysis of feed-water :
Permanent. Temporary.
J. Average steam pressure, lbs. per sq. in. :
Gauge. Absolute.
K. Temperature of saturation of steam.
L. Temperature of superheated steam.
M. Steam or power used as auxiliary to the production of steam.
N. Wetness of steam.

4. TABULATED RESULTS.

A. Duration of trial hours.
B. Lbs. of water evaporated per lb. of fuel as fired.

C. Equivalent lbs. water from and at 212° F. evaporated per lb. fuel as fired.

D. Equivalent lbs. water from and at 212° F. evaporated per 1,000,000 B.Th.U.

E. Efficiency of plant:
  - (*a*) Net working thermal efficiency of plant after deducting steam or power used auxiliary to the production of steam, corresponding to 4·2 per cent. of the total steam production.
  - (*b*) Efficiency of boilers only.
  - (*c*) Efficiency of economizers only.
  - (*d*) Efficiency of superheaters only.

F. Cost in fuel to evaporate 1,000 galls. of water (pence).

B. *Long Check Test.*

A. Duration.

B. Quality of fuel used.

C. Amount of fuel used.

D. Analysis of fuel used.
B.Th.U.
Ash. Water.

E. Water evaporated.

F. Water evaporated per lb. of fuel.

G. Cost in coal to evaporate 1,000 galls. of water (pence).

The method recommended of carrying out the test is as follows—

1. DAY'S TEST.

As regards the general description of the boiler plant, this will require practically no explanation, and it is very convenient to summarize this information as shown.

2. (A) *Weight of Fuel used.* The weight of fuel used on the trial can be determined by any convenient method. The firehole must be cleared out for the test, or else the existing coal covered with sacks or boards, so that there is no possibility of any fuel being used other than the weighed fuel. The fuel can be weighed in sacks, each sack of a standard weight, say, 2 cwts., and the number of sacks counted. Another method is to determine the net weight of a number of barrows and to calibrate each barrow to a fixed weight of coal (2–3 cwts.), as it is run over a scale in the firehole. A very convenient method is to weigh either horse carts or railway wagons direct, and, after emptying, carefully determine the tare of the empty vehicle. This method is quite accurate, as large weighing machines in good condition will weigh to 10 lbs. If an automatic weighing machine is attached, this can be tested with a small test load and then taken as accurate.

The ashes can be weighed if thought necessary.

The trial must be started up at the time agreed upon, with a certain thickness of fire, and finished up at the agreed time with the same thickness of fires, so that the weight of coal put into the fires will represent the actual weight of the coal burnt. In practice, this can be done quite easily.

(B) *Sampling the Fuel.* The very greatest care must be taken to secure an average sample of the fuel, since the calculations for the efficiency of the plant depend on the average heating value of the fuel. It is not an easy matter to take a true average sample of a fuel, and many boiler trials have been of little use because of this.

The best method is to take a good average sample of the fuel at least every ½ hr., and to put these samples in a box until the end of the trial. A good plan is to have two separate boxes, one for the samples taken at the hour by the clock and one for the ½ hr. by the clock. At the end of the trial, we have, therefore, two large boxes of fuel, each containing, say, 3 or 4 cwts. The boxes must be carefully placed in a position where the fuel, if wet, will not tend to dry during the day.

At the end of the trial, the contents of both boxes must be emptied separately on the floor, and the whole sample broken up with hand hammers and thoroughly mixed with a shovel. The sample is then spread out on the floor, "quartered," the quarter again "quartered," and so on until a final sample of about 4–5 lbs. is taken in a sealed tin. The remainder of the fuel is then used at the finish of the trial, or weighed separately and the weight deducted. In this way is obtained two entirely separate average samples of the coal used. These samples are then analysed separately for the heating value, and percentage of ash and water (as described on pp. 607–612), and an average taken for the figures of the two samples.

If the fuel is of bad quality or very mixed in character, the number of samples taken must be increased. If the plant is a large one, it is better to take samples separately for, say, each 20,000 lbs. tubular boiler, or each two or three "Lancashire" boilers.

(C) *Draught.* The draught in inches gauge must be taken every ½ hr. with an ordinary "U"-tube draught gauge, and the readings averaged, stating also the maximum and minimum. For "Lancashire" or similar boilers, the main positions are the side flues, before and after the economizer, and the chimney base. For tubular boilers, the exit of the flue gases, after the economizers, and the chimney base. Any additional positions will depend on the particular circumstances. With induced draught, one position can be in the main flue near the fan inlet.

A convenient method is to use ½ in. W.I. pipe with a rubber stopper connected by thin rubber tube to the draught gauge, which can be hung up for convenient reading.

Continuous recording draught gauges are, of course, an improvement, but are not really necessary.

(D) *Black Smoke.* This should be observed more or less continuously during the trial, and the results written down.

(E) *Temperature of Flue Gases.* The temperature of the flue gases must be taken every ½ hr. with an accurate type of pyrometer and the readings averaged, stating also the maximum and minimum. The main positions are before and after the economizer, but additional positions can, of course, be taken depending on the particular circumstances.

(F) *CO_2 in Flue Gases.* An automatic combustion recorder should be fitted on the plant and a continuous record of the CO_2 obtained. The ordinary practice of using a hand analysing machine such as an "Orsat" is useless, as only about one analysis can be completed, say, every 15 mins. The automatic recorder will give about twenty analyses per hour, and show a proper indication of the rise and fall of the percentage of CO_2. The only essential position for sampling the gases is as near the furnace as possible, with a "Lancashire" or similar boiler in the side flues or downtake, and with a tubular boiler through the side walls of the brickwork over the back of the furnace. Other positions can also be taken depending on circumstances, especially to detect leaky brickwork.

(G) *Flue Gas Analysis.* A further analysis of the flue gases must be made by taking very slowly a continuous sample of the flue gases by means of a large bottle or other apparatus. At the end of the trial, the bottle of gas, representing an average for the whole trial, can be taken to the laboratory and analysed at leisure for percentage of CO_2, oxygen, CO, and nitrogen, etc.

3. (A) WATER EVAPORATED.

The amount of water evaporated can be determined either by direct weighing in tanks, or by using a water meter.

The standard method of weighing the water in tanks is usually applied on the following general lines, using three tanks. Two graduated tanks BB are calibrated accurately to a certain level for a convenient weight of water, say, 5,000 lbs.; or the two tanks can be placed bodily each on a separate weighing machine, and feed-water run in until the weight is attained. By whichever method is used, a known weight of water in BB is then run down as required into the tank C. As each tank B is emptied, a note is taken, and at the end of the trial the number of tanks gives the weight of the water.

The tank C is connected direct to the boiler feed pump. The trial is started by filling C to a marked level with unweighed water and, when the trial commences, the contents of tanks BB are added as required. At the finish of the trial, the level in C must be adjusted to be the same as at the commencement by adding a small amount of weighed water.

The trial must be commenced by marking carefully the level of the water in the gauge glass of each boiler. Towards the end of the trial the boiler feed pump and the check valves must be gradually adjusted so that the level is about the same in each boiler as at the commencement. During the last half-hour of the trial, a final adjustment must be made with a view to getting all the levels exact as at the commencement. In practice, on most boiler plants this can be done quite easily. In any case, the levels must be carefully measured at the end of the trial and any difference calculated for, and allowance made in the total weight of water used. It is a little difficult to read the exact level in a gauge glass, as the water in the boiler has a rising and falling motion, but with care the error is not great.

It is very convenient to measure the gauge glass levels in each boiler every half-hour. By calculation from the width of the boiler at the point of the marked gauge glass, an approximate figure can be obtained for the exact evaporation each half hour. The figure is quite near enough to determine the fluctuation in the demand for steam for all practical purposes.

(B) *Temperature of Feed Water.* The temperature of the feed-water must be taken every half-hour before and after the economizers, and the readings averaged, stating also the maximum and minimum. This can be done with ordinary mercurial thermometers, or with pyrometers. Care must be taken that the real temperatures are taken. For example, if cold heated by exhaust steam, the real inlet temperature from the point of view of the boiler plant efficiency is after the heater, because the heat from exhaust steam is external to the boiler plant.

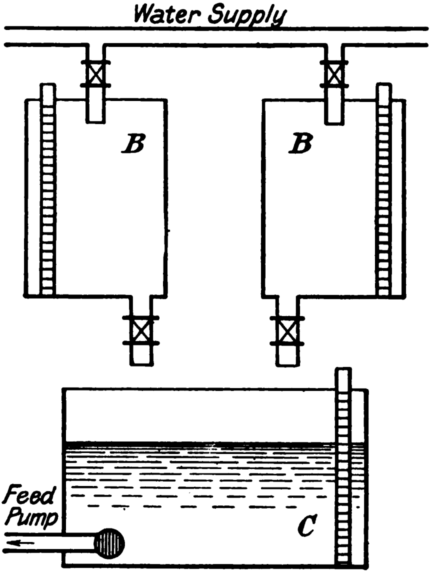


FIG. 37.—MEASUREMENT OF WATER EVAPORATED.

(C) *Analysis of Feed-water.* A good average sample of the feed-water must be taken during the test. The best method is to take a small sample, say, every half-hour and mix all the samples together at the end of the day.

(D) *Steam Pressure.* The steam pressure as usual must be read every half-hour and the figures averaged for the whole trial, stating also the maximum and minimum. If the pressure is not kept up easily, it is as well to take the readings oftener, say, every quarter-hour or even every 10 mins. An automatic steam pressure gauge is really a great advantage, as it gives a continuous record about which there can be no dispute.

(E) *Temperature of Superheated Steam.* This also must be taken as usual every half-hour with an ordinary mercurial thermometer, or special nitrogen-filled thermometer. The results are then averaged as before, stating the maximum and minimum. The improved system of pyrometer readings as applied at the same time to feed-water, flue gases, etc., is, of course, very convenient.

(F) *Steam or Power used as Auxiliary to the Production of Steam.* This is of the very greatest importance in determining the real net working efficiency of a boiler plant. It will be obvious that the efficiency must be calculated from the actual amount of steam available for useful work away from the boiler plant. Thus if a boiler is evaporating 10,000 lbs. of water per hour, and of the steam produced 1,000 lbs. per hour is blown away in the form of steam jets under the fire-bars, the real amount of steam produced is 9,000 lbs. per hour and not 10,000 lbs.

The basis of calculation must be the simplest plant, that is hand-firing with chimney draught only. The only auxiliary steam used will then be for the boiler feed pump and the very small engine for driving economizer scrapers. This amount is very small, and for practical purposes it can be ignored as common to almost all boiler plants.

Any extra steam or power used must be carefully determined and allowed for in the calculations. Thus—

(1) *Mechanical Stokers.* The h.p. taken to drive the stokers must be determined, and the amount of steam required must be deducted from the gross evaporation. As the power is small, this can be calculated from the particulars of the engine, revolutions, stroke, pressure, etc. If a motor is used, the method is simple.

(2) *Steam Nozzles.* As already described on pages 656–672, there are numerous forms of mechanical stokers and of special hand-fired furnaces using steam jets or nozzles. A careful observation must be made, spread over the week's check test as to how these nozzles are worked. In general, the method is to have all the steam supply valves full open all the time. In some cases, however, the firemen are more particular and only open the nozzles full at certain periods of the day, such as when cleaning out. For the rest of the day, the valves may be, say, ½ turn open, and so on. A record must be taken so that when the determination is carried out, the nozzles can be tested just as they are used in actual practice. When this

has been decided, the determination can be carried out. The apparatus suitable for this purpose has already been described on page 685. It must be rigged up in the firehole next to the boilers. The whole of the steam jet apparatus is then taken off one boiler and inserted bodily in the closed cylinder of the apparatus. This is then coupled up to the original jet pipe in front of the boiler, and the steam supply valve adjusted exactly the same as if the boiler was working. The fires can be practically banked up and the steam pressure in the boiler kept normal. The steam-jet apparatus is then working exactly as if it was under the boilers. The steam is condensed as already described and weighed off as water into buckets. The best method is to start up the whole apparatus, and let the steam jets work for some time, say, quarter of an hour, until everything is normal. A weighed bucket is then placed under the stream of condensed water at a stated time, and a series of half-hourly determinations carried out. Usually, four or five are ample, but if the figures are not more or less the same, more determinations can be carried out until a good average is obtained. In the same way, other determinations can be carried out at different steam pressures and with the steam supply valve in different positions, depending on the special conditions.

In this way, a good average figure can be obtained for the amount of steam used per hour on the plant, which has to be deducted from the gross evaporation. A simpler method is to blow the steam bodily into a weighed amount of cold water for a certain time until the water begins to get very hot. The water is then weighed again, and the increase in weight represents the steam used by the nozzles for the time. This method is not so accurate as that already described.

Other methods can, of course, be adopted, especially if the plant is a large one. Thus a steam meter can be fitted on the main steam pipe supplying all the nozzles.

If the plant is very large, undoubtedly the best method is to use one or two boilers only for producing the steam required for the nozzles of the whole plant. A water meter can be fitted on these boilers only, and the weight of the coal used also taken separately from the rest of the plant. The percentage of steam used by the nozzles is then plainly indicated without any dispute in terms not only of water evaporated, but in actual weight of coal as well.

(3) *Mechanical Draught*, that is the steam or power used by the engines or motors used to drive the fans. With motors, there is, of course, no difficulty. With engines, these can be indicated if large. For small engines, a good figure can be obtained by calculation from the dimensions of the engine revs. per minute, and steam pressure; whilst the figure for the lbs. of steam per i.h.p. can be taken from some analagous type of engine. Thus the ordinary small high-speed non-condensing fan engines take an average figure of 35 lbs. steam per i.h.p., and so on.

(G) *Wetness of the Steam.* Ordinary saturated steam as produced from a boiler plant without superheaters generally contains more or less moisture in the form of mist, say, 1–4 per cent. This is recorded on the water meter as steam, and tends therefore to make the efficiency of the plant too high.

In practice, it is a very difficult matter to get an average sample of steam and to determine the amount of moisture present. To obtain a sample, a perforated pipe is inserted into the main steam pipe so that a sample is drawn from every portion of the pipe. One method of determining the moisture is to blow the steam into cold water. The pressure is observed accurately, and the corresponding sensible and latent heat obtained from the steam tables.

The weight of the cold water was observed at the start, together with the temperature. Also the weight and temperature of the water at the finish. By calculation, the amount of moisture in the steam can be calculated because of the missing heat, since the moisture in the steam has no latent heat. In Carpenter's separating calorimeter, the steam flows very slowly through the apparatus, so that the moisture in the steam deposits, and only steam passes on. The Author is of the opinion that for ordinary practical purposes it is not necessary to determine the dryness factor if reasonable precautions are taken.

(B) *Long Check Test.* It is most advisable to carry out a check test of, say, 1 week complete, or a number of such weeks if thought necessary. This is not only a check on the day's trial, but included all the week-end losses and any night losses as well, so that it is really necessary to a proper record of the working of the plant.

The essential requirements are the total weights of water and fuel used, and the analysis of the fuel. The methods used are the same as for the day's trial, but it will generally be sufficient to take a sample of coal, say, every hour for each day and mix these samples at the end of the day, giving these six final samples in tins for analysis, in addition to the two tins for the day's test. As already stated, too much care cannot be taken with fuel sampling, and the method of taking the samples must be varied to suit the particular conditions of the plant. Also, it is often of advantage to take separately the records of the coal and water used for each day shift as compared with each night shift during the week. For example, it often happens that when a boiler plant is working with a reduced load at night, the efficiency is lower than during the day when full load is on and better attention given. Here, again, the tests must be adopted to suit the particular conditions.

COMPLETE METHOD OF CALCULATING THE RESULTS OF A BOILER TEST AND MAKING A REPORT

The results of the data for the calculation of the efficiency, etc., from a typical test as described, are as follows, taking the actual figures of an average specimen test—

1. General Description of the Boiler Plant—

A. "Lancashire." C. 30′ × 8′ E. Hand.
B. 4. D. 153·42 sq. ft. F. Patent steam-jet fire-bars.
G. 4 steam-jets (⅛ in. hole) on each furnace, that is 8 jets per boiler.
H. 320.
8. 9 ft. Straight line.
I. 185 ft. 6 ft. diameter. 8 ft. diameter.
J. Chimney (in addition to steam-jets).
K. 4 superheaters (in downtake as usual).
L. Double-acting vertical ("Cameron" type).
M. No injector.
N. Well water after passing through main engine condenser, and treatment in softening plant.
O. 60 lbs. P. 30 lbs.
Q. 2 on each shift (8 hrs.).
Average earnings about £4 5s. per week per man.
R. 1 on each shift (8 hrs.).
Average earnings about £3 10s. per week per man.
S. 19.
T. This plant is not coupled up to any other boiler plant.

2. Particulars Relating to the Burning of Fuel—

A. Good quality slack.
B. 39s. 10½d. delivered in firehole.
C. B.Th.U., 12,826; ash, 12·9 per cent.; water, 2·1 per cent.
D. 24640·0 lbs. (11 tons 0 cwts. 0 qrs.).
E. 717·9 lbs.
F. 18·7 lbs.
G. 0·59 in. W.G. (average of 17 readings).
H. 0·22 in. W.G. (average of 17 readings).
I. Very bad at times.
J. 590° F., average of 16 readings; max. 630° F., min. 540° F.
K. 446° F., average of 16 readings; max. 510° F., min. 400° F.
L. 8·0 per cent. (average of 20 tests per hour for one week).
M. 8·0 per cent. CO_2 (carbon dioxide). N. 79·9 (nitrogen).
0·2 per cent. CO (carbon monoxide). O. 11·9 (oxygen).

3. Particulars Relating to Steam Production—

A. 163362·0 lbs. Exact { On volume meter, 16578·0 galls. at 133° F., which corresponds to 16336·2 galls. or 163362·0 lbs. where 1 gall. equals 10·0 lbs.).

	Time.	*Evaporation per boiler per hour.*
B.	8. 0– 8.30 a.m.	384
	8.30– 9	470
	9. 0– 9.30	530
	9.30–10. 0	420
	10. 0–10.30	464

Time.	*Evaporation per boiler per hour.*
10.30–11. 0	580
11. 0–11.30	350
11.30–12. 0 noon	324
12. 0 noon–12.30 p.m.	450
12.30 p.m.–1.0 p.m.	440
1. 0– 1.30	534
1.30 p.m.–2. 0 p.m.	624
2. 0 „ –2.30 „	464
2.30 „ –3. 0 „	434
3. 0 „ –3.30 „	584
3.30 „ –4. 0 „	484
4. 0 „ –4.30 „	540

C. 475·7 galls. D. 12·4 galls.
E. Calibrated hot-water pressure meter.
F. 133° F., average of 15 readings; max. 154° F., min. 111° F.
G. 260° F., average of 15 readings; max. 280° F., min. 232° F.
H. 13·7 per cent.
I. 5° permanent hardness; 2° temporary hardness.
J. 40·0 lbs. per sq. in. Average of 17 readings.
55·0 lbs. per sq. in. Max. 48 lbs., min. 35 lbs.
K. 287° F. L. 385° F.
M. Average of 3 determinations of half-hour each gives 199·8 lbs. of steam per boiler per hour, that is, 799·2 lbs. of steam on the plant of 4 boilers per hour, corresponding to 4·2 per cent. of the steam production.

4. TABULATED RESULTS.

A. 8·58 hrs. (8 a.m.–4.35 p.m.).
B. 6·62 lbs. exact.
6·72 lbs. on volume meter with water at 133° F.
C. 7·32 lbs. E. (*a*) 56·22 per cent.
D. 570·7 lbs. (*b*) 49·29 „
(*c*) 6·77 „
(*d*) 2·62 „
F. 322·2 pence exact per 1,000 gallons.
317·5 pence on volume meter with water at 133° F.

(B) *Long Check Test.*

A. 168·50 hrs. (one week).
B. Good quality slack (39s. 10½d.) delivered.
C. 188 tons 17 cwts. 0 qrs.
D. 275600·0 galls. (at 130° F., approx.).
E. 6·31 lbs. F. 324·2 pence per 1,000 gallons.

1. WEIGHT OF WATER.

If a water-meter is used which is a volume measurer, the record given by the meter is the volume of water and not the true weight, and must be corrected accordingly.

The table for the density and relative volume of water as is given in page 753.

Thus in the trial, the actual figures on the water-meter are

16578·0 galls. of water at 133° F. Since the water is warm, it will have expanded, and the density is less; and, therefore, this does not correspond to 165,780 lbs. of water (1 gall. = 10·0 lbs.), but to some less amount.

On referring to the table on page 753, it will be seen that at 133° F. the water will have expanded in the ratio of approx. 1·0148 : 1·0000, so that the true weight of the water is 165,780 lbs. divided by 1·0148 equals 163,362 lbs. exact.

In this way, the correct weight of water from the record of a volume meter can be easily obtained for very accurate calculations. For continuous weekly records, it is not necessary to make this correction.

2. Equivalent Evaporation from and at 212° F.

In giving the results of a boiler test, it is not of much use in simply stating the lbs. of water evaporated per lb. of coal.

Important factors are obviously the temperature of the feed-water, the pressure of the boiler, and the heating value of the coal. It is customary in engineering to get over the first two objections to also express the figures as the "equivalent evaporation from and at 212° F. per lb. of coal"; that is to say, what the evaporation of water per lb. of coal *would be* if the temperature of the feed-water was 212° F., and the pressure of the steam was simply atmospheric (i.e. zero on the steam gauge), and the total heat only 966·1 units.

In the trial given, the figure for the evaporation of water per lb. of coal as given by simple division is 6·62 lbs.

As seen from the steam tables, the total heat saturated steam at 55·0 lbs. absolute and 133° F. is 1069·2. If now the total heat was only 966·1, the figure of 6·62 lbs. would be better in the ratio of $\frac{6{\cdot}62 \times 1069{\cdot}2}{966{\cdot}1} = 7{\cdot}32$ lbs. water per lb. of coal.

The figure can conveniently be obtained by means of the table shown on pages 734 and 735.

Thus from this table superheat does not enter into the calculation.

3. Equivalent Evaporation from and at 212° F. per 10,000,000 B.Th.U.

The above figure "from and at 212° F." is still very unscientific, because no notice is taken of the heating value of the fuel used. The Author proposes as a much more suitable figure the evaporation "from and at 212° F." from a fixed amount of heat, say 10,000,000 B.Th.U.

As seen, the figure "from and at 212° F." is 7·32, that is 1 lb. of coal, 12,826 B.Th.U. evaporated 7·32 lbs. water "from and at 212° F." Therefore 10,000,000 B.Th.U. would evaporate

$$\frac{7{\cdot}32 \times 10{,}000{,}000}{12{,}826} = 570{\cdot}7 \text{ lbs.}$$

4. Efficiency of Plant.

The efficiency of a boiler plant is the ratio between the amount of heat put into the plant in fuel, and the amount of

FACTORS OF

FACTORS FOR ASCERTAINING THE EQUIVALENT VARYING PRESSURES AND

Temperature of Feed Water. °Fahr.	BOILER PRESSURES (GAUGE)									
	65	70	75	80	85	90	95	100	105	110
40°	1·210	1·211	1·213	1 214	1·215	1·216	1·218	1·219	1·220	1·221
45°	1·204	1·205	1·207	1·208	1·209	1·210	1·212	1·213	1·214	1·215
50°	1·199	1·200	1·202	1·203	1·204	1·205	1·207	1·208	1·209	1·210
55°	1·194	1·195	1·197	1·198	1·199	1·200	1·202	1·203	1·204	1·205
60°	1·189	1·190	1·192	1·193	1·194	1·195	1·197	1·198	1·199	1·200
65°	1·184	1·185	1·187	1·188	1·189	1·190	1·192	1·193	1·194	1·195
70°	1·179	1·180	1·182	1·183	1·184	1·185	1·187	1·188	1·189	1·190
75°	1·174	1·175	1·177	1·178	1·179	1·180	1·182	1·183	1·184	1·185
80°	1·168	1·169	1·171	1·172	1·173	1·174	1·176	1·177	1·178	1·179
85°	1·163	1·164	1·166	1·167	1·168	1·169	1·171	1·172	1·173	1·174
90°	1·158	1·159	1·161	1·162	1·163	1·164	1·166	1·167	1·168	1·169
95°	1·153	1·154	1·156	1·157	1·158	1·159	1·161	1·162	1·163	1·164
100°	1·148	1·149	1·151	1·152	1·153	1·154	1·156	1·157	1·158	1·159
105°	1·142	1·143	1·145	1·146	1·147	1·148	1·150	1·151	1·152	1·153
110°	1·137	1·138	1·140	1·141	1·142	1·143	1·145	1·146	1·147	1·148
115°	1·132	1·133	1·135	1·136	1·137	1·138	1·140	1·141	1·142	1·143
120°	1·127	1·128	1·130	1·131	1·132	1·133	1·135	1 136	1·137	1·138
125°	1·122	1·123	1·125	1·126	1·127	1·128	1·130	1·131	1·132	1·133
130°	1·116	1·117	1·119	1·120	1·121	1·122	1·124	1·125	1·126	1·127
135°	1·111	1·112	1·114	1·115	1·116	1·117	1·119	1·120	1·121	1·122
140°	1·106	1·107	1·109	1·110	1·111	1·112	1·114	1·115	1·116	1·117
145°	1·101	1·102	1·104	1·105	1·106	1·107	1·109	1·110	1·111	1·112
150°	1·096	1·097	1·099	1·100	1·101	1·102	1·104	1·105	1·106	1·107
155°	1·090	1·091	1·093	1·094	1·095	1·096	1·098	1·099	1·100	1·101
160°	1·085	1·086	1·088	1·089	1·090	1·091	1·093	1·094	1·095	1·096
165°	1·080	1·081	1·083	1·084	1·085	1·086	1·088	1·089	1·090	1·091
170°	1·075	1·076	1·078	1·079	1·080	1·081	1·083	1·084	1·085	1·086
175°	1·070	1·071	1·073	1·074	1·075	1·076	1·078	1·079	1·080	1·081
180°	1·064	1·065	1·067	1·068	1·069	1·070	1·072	1·073	1·074	1·075
185°	1·059	1·060	1·062	1·063	1·064	1·065	1·067	1·068	1·069	1·070
190°	1·054	1·055	1·057	1·058	1·059	1·060	1·062	1·063	1·064	1·065
195°	1·049	1·050	1·052	1·053	1 054	1 055	1·057	1·058	1 059	1 060
200°	1·044	1·045	1·047	1·048	1·049	1·050	1·052	1·053	1·054	1·055
205°	1·039	1·040	1·042	1·043	1·044	1·045	1·047	1·048	1·049	1·050
210°	1·034	1·035	1·037	1·038	1·039	1·040	1·042	1·043	1·044	1·045
215°	1·029	1·030	1·032	1·033	1·034	1·035	1·037	1·038	1·039	1·040
220°	1·024	1·025	1·027	1·028	1·029	1·030	1·032	1·033	1·034	1·035
	TEMPERATURE OF									
	312	316	320	324	328	331	334	338	341	344

Allowance for Superheat—For every

EVAPORATION

Rate of Evaporation from and at 212° Fahr. for Temperatures of Feed-Water

Lbs. per Square Inch.

115	120	125	130	140	150	160	170	180	190	200	Temperature of Feed Water °Fahr.
1·222	1·223	1·224	1·225	1·226	1·228	1·229	1·230	1·231	1·232	1·233	40°
1·216	1·217	1·218	1·219	1·220	1·222	1·223	1·224	1·225	1·226	1·227	45°
1·211	1·212	1·213	1·214	1·215	1·217	1·218	1·219	1·220	1·221	1·222	50°
1·206	1·207	1·208	1·209	1·210	1·212	1·213	1·214	1·215	1·216	1·217	55°
1·201	1·202	1·203	1·204	1·205	1·207	1·208	1·209	1·210	1·211	1·212	60°
1·196	1·197	1·198	1·199	1·200	1·202	1·203	1·204	1·205	1·206	1·207	65°
1·191	1·192	1·193	1·194	1·195	1·197	1·198	1·199	1·200	1·201	1·202	70°
1·186	1·187	1·188	1·189	1·190	1·192	1·193	1·194	1·195	1·196	1·197	75°
1·180	1·181	1·182	1·183	1·184	1·186	1·187	1·188	1·189	1·190	1·191	80°
1·175	1·176	1·177	1·178	1·179	1·181	1·182	1·183	1·184	1·185	1·186	85°
1·170	1·171	1·172	1·173	1·174	1·176	1·177	1·178	1·179	1·180	1·181	90°
1·165	1·166	1·167	1·168	1·169	1·171	1·172	1·173	1·174	1·175	1·176	95°
1·160	1·161	1·162	1·163	1·164	1·166	1·167	1·168	1·169	1·170	1·171	100°
1·154	1·155	1·156	1·157	1·158	1·160	1·161	1·162	1·163	1·164	1·165	105°
1·149	1·150	1·151	1·152	1·153	1·155	1·156	1·157	1·158	1·159	1·160	110°
1·144	1·145	1·146	1·147	1·148	1·150	1·151	1·152	1·153	1·154	1·155	115°
1·139	1·140	1·141	1·142	1·143	1·145	1·146	1·147	1·148	1·149	1·150	120°
1·134	1·135	1·136	1·137	1·138	1·140	1·141	1·142	1·143	1·144	1·145	125°
1·128	1·129	1·130	1·131	1·132	1·134	1·135	1·136	1·137	1·138	1·139	130°
1·123	1·124	1·125	1·126	1·127	1·129	1·130	1·131	1·132	1·133	1·134	135°
1·118	1·119	1·120	1·121	1·122	1·124	1·125	1·126	1·127	1·128	1·129	140°
1·113	1·114	1·115	1·116	1·117	1·119	1·120	1·121	1·122	1·123	1·124	145°
1·108	1·109	1·110	1·111	1·112	1·114	1·115	1·116	1·117	1·118	1·119	150°
1·102	1·103	1·104	1·105	1·106	1·108	1·109	1·110	1·111	1·112	1·113	155°
1·097	1·098	1·099	1·100	1·101	1·103	1·104	1·105	1·106	1·107	1·108	160°
1·092	1·093	1·094	1·095	1·096	1·098	1·099	1·100	1·101	1·102	1·103	165°
1·087	1·088	1·089	1·090	1·091	1·093	1·094	1·095	1·096	1·097	1·098	170°
1·082	1·083	1·084	1.085	1·086	1·088	1·089	1·090	1·091	1·092	1·093	175°
1·076	1·077	1·078	1·079	1·080	1·082	1·083	1·084	1·085	1·086	1·087	180°
1·071	1·072	1·073	1·074	1·075	1·077	1·078	1·079	1·080	1·081	1·082	185°
1·066	1·067	1·068	1·069	1·070	1·072	1·073	1·074	1·075	1·076	1·077	190°
1·061	1·062	1·063	1·064	1·065	1·067	1·068	1·069	1·070	1·071	1·072	195°
1·056	1·057	1·058	1·059	1·060	1·062	1·063	1·064	1·065	1·066	1·067	200°
1·051	1·052	1·053	1·054	1·055	1·057	1·058	1·059	1·060	1·061	1·062	205°
1·046	1·047	1·048	1·049	1·050	1·052	1·053	1·054	1·055	1·056	1·057	210°
1·041	1·042	1·043	1·044	1·045	1·047	1·048	1·049	1·050	1·051	1·052	215°
1·036	1·037	1·038	1·039	1·040	1·042	1·043	1·044	1·045	1·046	1·047	220°

Saturated Steam. °Fahr.

115	120	125	130	140	150	160	170	180	190	200
347	350	353	356	361	366	371	375	380	384	388

10° Superheat add ·00548 to Factor.

this heat that is used to produce useful steam. Thus, 60 per cent. efficiency means that for every 100 lbs. of coal put into the fires, 60·0 lbs. is used to produce useful steam, and the other 40 lbs. is wasted.

In calculating this efficiency, we must first determine from the steam tables the theoretical amount of heat required for 1 lb. of water.

The temperature of the inlet water is 133° F. Since the steam tables are based on 32° F., the total heat will therefore, be less on this account.

As already stated on page 613, it is not sufficiently accurate to subtract 133° F. from 212° F., equals 79, and then subtract 79 units of heat. The more correct formula is—

$$\begin{aligned} S &= 1\cdot017t - 35 \\ &= 1\cdot017 \times 133^\circ - 25 \\ &= 135\cdot3 - 35 \\ &= 100\cdot3 \end{aligned}$$

From 32° F., at 55·0 lbs. per sq. inch absolute, the total heat of saturated steam, as seen from the tables, is 1169·5. From 133° F., therefore, the heat is 1169·5 – 1069·2.

In calculating the efficiency due to the economizers only, we shall also need the total heat of steam from 260° F., the temperature of the feed-water after the economizers.

This is obtained exactly in the same way—

$$\begin{aligned} S &= 1\cdot017t - 35 \\ &= 1\cdot017 \times 260 - 35 \\ &= 264\cdot4 - 35 \\ &= 229\cdot4 \end{aligned}$$

The total heat from 260° F. is, therefore, $1169\cdot5 - 229\cdot4 = 940\cdot1$. Further, also, the steam is superheated, being 385° F. As seen from the steam tables, the temperature of the boiling point of water at 55 lbs. absolute is 287·0° F., so that the amount of superheat is 385°– 287° equals 98°.

The amount of heat units necessary for this superheat is determined from the Mollier diagram curve. On consulting this curve, it will be seen that the specific heat of superheated steam at 55 lbs. absolute and 385° F. is approx. 0·505.

The heat units are, therefore, $98 \times 0\cdot505 = 49\cdot49$ units.

This must, therefore, be added on to the total heat of saturated steam, and we have the total heats of steam at 55 lbs. absolute as follows—

1. Non-superheat—
(*a*) From 133° F. = 1089·2 (*b*) From 260° F. = 940·1

2. Superheated to 385° F.—
(*a*) From 133° F. = 1118·7 (*b*) From 260° F. = 989·6

We now turn to the actual amount of heat taken on the plant to raise 1 lb. of water at 133° F. to steam at 55 lbs. absolute and 385° F. temp. due to superheating.

163362·0 lbs. water are evaporated by 46240, lbs. coal. Therefore 1 lb. water is evaporated by 0·15083 lbs. coal.

Now the heating value of the coal is 12,826 B.Th.U. per lb., so that 1 lb. of water is evaporated by 193,454 B.Th.U.

(12,826 × 0·15083).

As seen, this is much more than the theoretical figure, so that the efficiency of the plant is much less than 100.

We now have—

1. *Non-superheated Steam*—

(a) Efficiency of plant (boilers and economizers) from 133° F. equals :

$$\frac{1069 \cdot 2 \times 100}{1934 \cdot 54} = 55 \cdot 26\%$$

(b) Efficiency of boilers only, that is, calculated from 260° F. and eliminating the economizers

$$\frac{940 \cdot 1 \times 100}{1934 \cdot 54} = 48 \cdot 59\%$$

(c) Efficiency of economizers only, by difference

$$55 \cdot 26 - 48 \cdot 59 = 6 \cdot 67\%$$

2. *Superheated Steam* (385° F.)—

(a) Efficiency of plant (boilers, economizers, and superheaters) equals :

$$\frac{1118 \cdot 7 \times 100}{1934 \cdot 54} = 57 \cdot 82\%$$

As above 1 (*a*) the efficiency of boilers and economizers is 55·26 per cent., the efficiency due to superheaters only is—

$$57 \cdot 82 - 55 \cdot 26 = 2 \cdot 56\%$$

(b) As an additional check, we have efficiency of boilers and superheaters only :

$$\frac{989 \cdot 6 \times 100}{1934 \cdot 54} = 51 \cdot 15\%$$

Therefore efficiency due to superheaters is 51·15% – 48·59% (1b.) = 2·56%.

Also to economizers :

$$57 \cdot 82\% \ (2a) - 51 \cdot 15\% = 6 \cdot 67\%$$

The separate figures for the efficiency are, therefore—

Boiler only..	48·59%
Economizer only..	6·67%
Superheater only	2·56%
Total	57·82%

This, however, is not the total net working efficiency, because of the complication due to steam used by the steam nozzles, namely, 4·2 per cent. of the production. If a boiler plant evaporates 100 lbs. of water to steam and blows 10 lbs. away under the bars, the real useful steam available for work at the stop valve of the boiler is 90 lbs., not 100 lbs.

In calculating the net working efficiency, therefore, this 10 lbs. has to be deducted.

As seen from the test, 4·2 per cent. of the steam produced is used by the jets, that is for every 100 lbs. evaporated only 95·8 lbs. is useful steam. We have seen that the total heat actually taken is—

(*a*) Non-superheat from 133° F. = 1069·2.

(*b*) Superheat to 385° F. from 133° F. = 1118·7.

Now, since only 95·8 per cent. of the production is useful steam, the real total heat taken in producing useful steam is really more than shown by the steam tables, namely

$$\frac{1934{\cdot}5 \times 100}{95{\cdot}8} = 2019{\cdot}3$$

The real net working efficiency of the plant complete is, therefore—

(*a*) Non-superheat as 1 (*a*)—

$$\frac{1069{\cdot}2 \times 100}{2019{\cdot}3} = 52{\cdot}94\%$$

(*b*) Superheat as 2 (*a*)—

$$\frac{1118{\cdot}7 \times 100}{2019{\cdot}3} = 55{\cdot}40\%$$

The final results, therefore, of this particular test can be expressed as follows—

(*a*) (4·2%) = 55·40%   (*c*) (4·2%) = 6·67%
(*b*) = 48·59%   (*d*) = 2·56%

5. SAVING DUE TO ECONOMIZER.

As we have seen, the economizers have an efficiency of 6·67 per cent., because they heat the feed-water from 133° F. to 260° F. The question of superheat does not enter into the calculation since it relates to the steam only, and not the water.

The boiler efficiency is 48·59 per cent., so that the efficiency of boilers and economizers only is 55·26 per cent. To express the figure for the economizers in percentage saving of the coal bill, if a total efficiency of 55·26 per cent. saves 6·67 per cent.,

then $\frac{100 \times 6{\cdot}67}{55{\cdot}26}$ = 12·0 per cent., the saving in the coal bill.

The saving can be conveniently determined by the table already given.

The Author has given this matter much attention, and has made a complete scientific investigation of nearly 500 boiler plants and has examined about 2,000 plants.

He published in *Engineering*, 12th July, 1918 ("Coal Saving by the Scientific Control of Steam Boiler Plants"), the results of the complete scientific investigation of 250 boiler plants in Great Britain carried out during the years 1910–1920.

These plants were situated in England, Scotland, Wales, and

Ireland; are thoroughly representative of the boiler plants of Great Britain; and include the following industries—

Bleaching
Brewing
Calico printing
Carpet manufacture
Chemical (general)
Collieries
Cotton manufacture
Dyeing
Dyeing and cleaning
Electric power stations
Engineering
Explosives
Flour milling
Food products
Hat manufacture
Hosiery manufacture
Hospitals
Hotels
Lace manufacture
Laundry
Leather manufacture
Linen manufacture
Paper manufacture
Potteries
Soap manufacture
Steel manufacture
Woollen manufacture

They comprised all sizes of plants with annual coal bills of from 750 tons to 70,000 tons, whilst the average working pressure varied from 12 lbs. to 200 lbs. per square inch.

The fuels used are of every possible description—from the highest qualities of washed nuts, beans peas, and fine slack, and large and small screened coal, down to inferior fuels of all qualities—including refuse coal mixed with shale, pit sweepings, washer settlings, etc., as well as coke and coke-breeze.

The figures for the whole 250 plants were averaged mathematically; that is to say, the 250 plants were regarded as one huge plant, the weight of coal used and water evaporated was added up, and all the various factors, such as heating value of the fuels, temperatures, draught, etc., etc., were obtained as a true average by laborious calculations. It would obviously not be a true average to simply add the figures and divide by 100, because of the different sizes and capacities of the individual plants.

The true average figures for these 250 plants were as follows—

(A) *Working Day's Test.*	*Approximately.*
Type of boiler	"Lancashire"
Number of boilers	4
Grate area	152·6 sq. ft.
Duration of test	9·43 hrs.
Price of fuel used	14s. 6d. per ton deliv'd.
Amount of fuel used	30,131·72 lbs.
Analysis of fuel used:	
British Thermal Units	11,822
Ash	11·5%
Fuel burnt per boiler per hour	798·8 lbs.
Fuel burnt per square foot of grate area per hour	20·9 lbs.
Water evaporated	197,776·0 lbs.
,, ,, per boiler per hr.	5,243 lbs.
Water evaporated per square foot of grate area per hour	137·4 lbs.
Water evaporated per lb. of fuel	6·56 lbs.

(A) *Working Day's Test.*	*Approximately.*
Equivalent evaporation from and at 212° F. per lb. of fuel	7·46 lbs.
Equivalent evaporation from and and at 212° F. per 1,000,000 B.Th.U.	631·0 lbs.
Temperature of feed-water before economizers	116° F.
Temperature of feed-water after economizers	193° F.
Percentage of fuel bill saved by economizers	7·1%
Draught in side flue	0·41 in. W.G.
Draught in main flue at exit of economizers or chimney-base	0·80 in. W.G.
Temperature of flue gases before economizers	598° F.
Temperature of flue gases after economizers	478° F.
Percentage of CO_2 in flue gases from side flue of boilers by means of combustion recorder	7·6%
Steam pressure :	
Lbs. per sq. inch (gauge)	89
,, ,, (absolute)	104
Temperature of saturation of steam	330·5° F.
,, of superheated steam	346·5° F.
Steam or power used as auxiliary to the production of steam	2·4%
Thermal efficiency :	
(A) Net working efficiency complete, after deducting 2·4% steam or power used as auxiliary to the production of steam	60·09%
(B) Boilers only	56·71%
(C) Economizers only	4·35%
(D) Superheaters only	0·51%
Cost in coal to evaporate 1,000 galls. of water :	
(*a*) 14s. 6d. (average price)	118·3d.
(*b*) 20s. 0d. (to-day's price)	163·2d.

(B) *Long Check Test.*

(One week of 7 days.)

Duration	167·5 hrs.
Price of fuel used	14s. 6d. per ton deliv'd
Amount of fuel used	128·25 tons
Water evaporated	184,435·0 galls.
,, ,, per lb. of fuel	6·42 lbs.
Cost in fuel to evaporate 1,000 galls. of water :	
(*a*) Fuel at 14s. 6d. (av. price)	120d.
(*b*) ,, 20. 0d. (to-day's ,,)	166·9d.

1918

These 250 plants had a total of 1,000 boilers, comprising 935 "Lancashire" boilers, 36 "water-tube" boilers, 17 "egg-ended" boilers, 6 "Cornish" boilers, 1 "Marine" boiler, 2 "vertical" boilers, and 3 "patent" boilers difficult to classify. The approximate grate length was 6 ft.; 190 of the plants (76 per cent.) were hand-fired and 60 plants (24 per cent.) mechanically-fired. The average evaporation calculated to standard 30′ × 8′ "Lancashire" boilers is about 600 galls. per boiler per hour, much less than is usually supposed. The average hardness of the feed-water was about 12 grs. per gall. total hardness; 155 of the plants were fitted with economizers, and on these plants only the saving was 11·4 per cent. of the coal bill; 95 plants had no economizers, so that the average for the whole 250 tests was only 7·1 per cent.; 123 of the plants were working with chimney-draught, 93 with steam-jet forced-draught, 32 with induced-draught, and 2 with (mechanical) forced-draught. The average figure for CO_2 was 7·6 per cent., and the plants were divided as follows—

	No. of Plants.	*Per Cent.*
1. Very good, over 12%	4	1·60
2. Good, 10–12%	17	6·80
3. Medium, 8–10%	64	25·60
4. Poor, 5–8%	144	57·66
5. Very bad, under 5%	21	8·40
Total	250	

That is, 66 per cent. of cases were below 8 per cent. The best record found was 14 per cent. average, and the worst 3·5 per cent. average.

As regards black smoke, the figures were—

	No. of Plants.	*Per Cent.*
1. Good: practically no smoke at all	65	26·0
2. Fairly good	62	24·8
3. Medium	75	30 0
4. Bad	43	17·2
5. Chronic; almost continuous black smoke	5	2·0
Total	250	

As regards steam pressures, all varieties of average working pressure were experienced from 12 lbs. (blow-off 30 lbs.) to

200 lbs. (blow-off 205 lbs.). The plants were divided as follows :

	No. of Plants.	*Per Cent.*
1. Under 60 lbs. average working pressure	45	18·0
2. Between 60 lbs. and 80 lbs. working pressure	71	28·4
3. Between 80 lbs. and 100 lbs. working pressure	55	22·0
4. Between 100 lbs. and 120 lbs. working pressure	35	14·0
5. Between 120 lbs. and 160 lbs. working pressure	37	14·8
6. Over 160 lbs. working pressure	7	2·8
Total	250	

Concerning superheaters, out of the 250 plants, 80 plants were fitted with superheaters, which includes 55 plants only partially fitted. Of the 80 plants fitted, the average temperature of steam was 316° F. before superheating, and 366° F. after, being 50° F. of superheat. The highest temperature of superheated steam test was 650° F. In the case of the 25 plants completely equipped with superheaters, the average amount of superheat was about 120° F.

The value of superheating is very little realized, either from the point of view of partial superheat of, say, 75° F., to avoid condensation losses in the steam-pipe circuit, or from that of complete superheating with a view to economical utilization of steam in the engines or turbines. As seen, the average of the whole 250 plants was only 16° F.

Coming to the steam or power used as auxiliary to the production of steam, as already seen, out of the 250 plants, 127 were using steam or power as an auxiliary to the production of steam, 93 being steam-jets under or over the bars, 32 as induced-draught, and 2 as forced-draught. Out of the 129 cases the average amount of steam so used was 4·7 per cent. of the production in the boilers, corresponding to 2·4 per cent. for the whole of the 250 plants.

With regard to induced-draught fans, the average is approximately 2½ per cent. to 3 per cent. of the production.

With steam-jets, the amount varies very considerably, being anything from 1 per cent. to 20 per cent. of the production, the average on the 93 plants being 5·6 per cent.

(*See also* page 685 with regard to the amount of steam used by hand-fired steam-jet furnaces ; and page 656 for mechanical stokers.)

With regard to the important factor of the net working efficiency, as seen, the average working efficiency for the whole 250 plants is 60·09 per cent., that is to say, for every 100 lbs.

of coal put into the boiler fires, 60·09 lbs. are being used to produce useful steam, the other 39·91 lbs. being wasted. The efficiency is split up into 56·71 per cent. for the boilers only, 4·35 per cent. for the economizers, and 0·51 per cent. for the superheaters, which, after allowing 2·4 per cent. used as auxiliary steam, gives a net working efficiency of 60·09 per cent.

Taking the whole 250 plants, the efficiencies were divided as follows—

	No. of Plants.	*Per Cent.*
1. Over 80%	2	0·8
2. 75%–80%	9	3·6
3. 70%–75%	13	5·2
4. 65%–70%	30	12·0
5. 60%–65%	44	17·6
6. 55%–60%	62	24·8
7. 50%–55%	47	18·8
8. Less than 50%	43	17·2
Total	250	

It will be noted that 61 per cent. of the plants are working with an efficiency of less than 60 per cent., and that only 9·6 per cent. of the plants exceed 70 per cent. efficiency.

Comparing one industry with another, the table shown on page 744 will be of interest.

The Author has also published in *Engineering*, 25th July and 1st August, 1919 (" Exact Data on the Running of Steam-boiler Plants: No. 2—The Performance of Colliery Boiler Plants "), the complete separate figures for the investigation of 100 colliery boiler plants.

The average net working efficiency for these 100 plants was 55·52 per cent., varying from a maximum of 71·84 per cent. to a minimum of 32·50 per cent.

Also in the *Transactions of the Institution of Mechanical Engineers* (paper read 19th March, 1920), the Author gave the complete results of the working of 80 " Lancashire " boiler plants mechanically-fired. The average net working efficiency was 59·0 per cent. varying from a maximum of 80·1 per cent. to a minimum of 40·1 per cent.

The Author is of the opinion that the average net working efficiency of the boiler plants of Great Britain, including boilers, economizers, and superheaters, is about 60–62 per cent.

For ordinary large " Lancashire " boiler plants with economizers and superheaters, the average figure is 60–65 per cent., the boilers only being about 55–56 per cent. Run on modern lines, the figure should be 75 per cent. net working efficiency, and under the most favourable conditions it is possible to get 80 per cent.

	75 Collieries.	20 Cotton Mills.	10 Lace Mills.	16 General Chemical Manufacturers.	22 Explosive Factories.	26 Dyers and Bleachers.	21 Woollen Mills.	9 Hosiery Mills.	21 Paper Mills.
Duration of trial . . . hrs.	8·4	11·15	9·14	7·98	17·66	·87	8·6	8·29	11·35
Amount of coal used. . . lbs.	41,851	28,773	13,897	11,939	210,741	17,744	17,462	15,251	37,558
Price of coal per ton (taken as) .	20s.	20s.	20s.	20s.	20s.	20s.	20s.	20s.	20s.
Analysis of coal used—									
British Thermal Units . . .	10,920	12,972	10,249	11,669	12,359	12,355	12,617	10,683	12,347
Ash . . . . . per cent.	16·5	8·4	12·4	11·2	9·67	8·5	11·0	9·5	10·6
Coal burnt per boiler per hour . lbs.	807·6	806·4	661·0	680·0	850·0	778·3	822·0	766·5	787·8
Water evaporated . . . lbs.	226,630	213,606	69,059	77,000	1,581,480	114,241	119,973	89,243	294,933
" " per lb. of coal lbs.	5·41	7·42	4·96	6·45	7·50	6·43	6·87	5·85	7·85
Equivalent evaporation from and at 212° F. per lb. of coal . . lbs.	5·8	8·54	5·70	7·28	8·78	7·39	7·88	6·72	8·83
Temp. of feed-water before economisers ° F.	163·0	110·0	96·0	121·0	89·5	98·0	109·0	105·0	133·0
Temp. of feed-water after economizers ° F.	—	250·5	126·0	199·0	183·0	207·0	237·0	186·0	261·0
Coal bill saved by economizers per cent.	—	12·8	2·75	7·30	8·4	9·9	11·7	7·4	11·9
CO_2 . . . . . per cent.	8·0	8·1	6·5	8·0	8·7	8·2	7·1	7·0	8·2
Steam pressure (gauge) . . lbs.	82·0	124·0	56·0	71·0	118·0	59·0	94·0	80·0	113·0
Temp. of saturation of steam . ° F.	325·0	352·0	304·0	317·0	349·0	307·0	333·9	323·9	345·9
Temp. of superheated steam . ° F.	—	415·0	—	357·0	401·0	320·0	345·4	346·6	412·8
Steam used as auxiliary . per cent	1·5	1·1	0·3	0·71	5·68	1·7	2·8	3·0	1·2
A. Net wk'ng efficiency after deducting steam used as auxiliary %	51·2	64·97	53·76	61·0	66·48	57·20	58·97	59·57	70·67
B. Boilers only	52·0	55·49	52·44	55·89	62·98	52·10	53·24	56·23	60·87
C. Economizers only . . .	nil	8·18	1·48	4·39	5·77	5·77	7·09	4·51	8·28
D. Superheaters only . . .	"	2·02	nil	1·15	1·73	0·31	0·34	0·66	2·38
Cost in coal to evaporate 1,000 galls. water per hour . . pence	197·8	144·3	215·6	166·1	142·4	166·4	155·9	183·1	136·4

Mechanical firing (*see* p. 656) is giving inferior results to hand-firing, but with modern methods of scientific control there is little difference.

With water-tube boilers fitted with economizers and super-heaters, the average figure is only about 70–72½ per cent. net working efficiency, the boiler only being, say, 62–65 per cent. Run on modern lines a water-tube boiler-plant will give 77½–80 per cent., and 82½ per cent. under the most favourable conditions.

With regard to small "Lancashire" and "Cornish" boiler plants without economizers and superheaters, on the usual lines the net working efficiency is about 50–55 per cent. The net working efficiency of small vertical boilers as usually worked is about 45 per cent.

The table given on pages 746 and 747 gives the average figures for different boiler plants as usually being obtained according to the Author, from *Engineering*, 10th and 17th Dec., 1920.

INSTRUMENTS AND APPLIANCES NECESSARY TO SCIENTIFIC CONTROL OF BOILER PLANT

I. Measurement of Boiler Feed-water (Hot-water Meters).

The old method of measuring feed-water by means of weighed tanks is obviously not suitable for the continuous weekly records which are necessary for maximum efficiency. Therefore, some form of water meter is required.

It is an extremely difficult matter to design a simple, reliable, and accurate apparatus to measure continuously, from week to week, boiler-feed water at varying temperatures from cold to nearly boiling; and which may also be muddy, gritty, oily, corrosive, and contain much scale-forming elements, and in which also the demand fluctuates considerably.

Hot-water meters can be divided into two general types: (*a*) non-pressure or open types; and (*b*) pressure or closed types.

Of the *non-pressure* types, we have—

(1) Automatic water-weighing machines—a direct development of the original tank system.

(2) Meters using "V" notches, weirs, and orifices.

Of the *pressure* types, we have—

(1) Meters of the piston or rotary type.

(2) "Venture" principle meters.

Non-pressure Type (1). The general principle of these meters is that a balanced tank of known dimensions is gradually filled with water from a supply pipe; and, as soon as the tank is filled to the required weight, a float or other mechanism releases a catch, and the tank tipples over into the feed-pump supply tank. At the same time, a recording mechanism with a train of wheels records the number of tanks tipped, and the weight is read off direct from the dials. They are supplied in a great variety of sizes, from about 20 lbs. up to 2½ tons per tank.

	CYLINDRICAL BOILER ("Lanc," etc.).		
	Average Plant, 85%.	Good Plant, 5%.	Bad Plant, 10%.
	Of Plants of Great Britain.		
Coal burnt per boiler per hour	lbs. 864·7	lbs. 1059·8	lbs. 755·2
Coal burnt per sq. ft. grate area per hr. ..	22·7	27·9	19·8
Water evap. per sq. ft. grate area per hr. ..	151·3	223·7	111·3
Water at 110° F. evap. per lb. coal	6·65	8·02	5·62
Water from and at 212° F., evap. per lb. coal.	7·59	9·28	6·39
Water from & at 212° F. per 10,000,000 B.th.u.	632·6	856·7	532·5
Temp. of inlet water ..	110° F.	110° F.	110° F.
Temp. of water after economizers	230° F.	335° F.	no econ.
Saving due to econs.	11·0%	20·4%	nil
Draught in chimney-base on fan inlet ..	0·75″W.G.	2·25″W.G.	0·50″W.G.
Temp. of gases leaving boiler	600° F.	650° F.	500° F.
Temp. of gases at chimney base or fan inlet	450° F.	310° F.	500° F.
Analysis of feed-water :			
Permanent hardness	9° F.	5° F.	12° F.
Temporary hardness	2° F.	0° F.	5° F.
% CO_2 in gases leaving the furnaces	7·5%	12·0%	5·0%
Aver. gauge pres. lbs. .	75	159	60
Temp. of saturated steam..	320·1° F.	350·2° F.	307·4° F.
Temp. of superheated steam..	nil	540° F.	nil
Steam or power used auxiliary to the production of steam	2·5%	2·5%	5·%
Efficiency :			
Net working ..	59·6%	57·7%	48·9%
Boilers only	54·4%	58·4%	51·5%
Economizers only	6·7%	15·0%	nil
Superheaters only	nil	6·3%	nil
Cost in coal to evap. 1,000 galls. of water *d.*	332·6	267·0	379·8
Coal bill for 20,000,000 galls. evap. based	£27,472	£22,748	£32,166

Water-tube Boiler.			Small Cylindrical Boiler Average.	Small Vertical Boiler Plant.	Egg-ended Boiler Average.
Average Plant, 85%.	Good Plant, 5%.	Bad Plant, 10%.			
Of Plants of Great Britain.					
2938·3	2859·9	2911·7	346·8	112·75	630·0
20·9	20·4	20·8	17·3		22·5
147·2	160·9	133·7	101·7		83·9
7·01	7·87	6·43	5·86	5·25	3·73
8·10	9·11	7·43	6·68	5·98	4·23
675·0 110° F.	759·2 110° F.	619·2 100° F.	556·6 110° F.	490·0 110° F.	352·5 110° F.
195° F. 7·6%	225° F. 10·5%	no econ. nil	no econ. nil	no econ. nil	no econ. nil
0·50″	0·65″	0·75″	0·50″	0·30″	1·00″
475° F.	450° F.	575° F.	590° F.	800° F.	850° F.
325° F.	300° F.	575° F.	590° F.	800° F.	850° F.
6° F. 2° F.	3° F. 0° F.	9° F. 2° F.	9° F. 2° F.	9° F. 2 F.	12° F. 5° F.
6·0% 155	12·5% 160	5·0% 150	5·0% 70	5·0% 70	3·75% 55
368·2° F.	370·5° F.	365·7° F.	316·1° F.	316·1° F.	302·8° F.
560° F.	650° F.	450° F.	450° F.	nil	nil
2·0%	1·5%	2·5%	nil	nil	nil
69·1% 60·3% 5·0% 5·2%	81·8% 65·7% 7·7% 9·6%	60·9% 59·8% nil 2·6%	53·8% 53·8% nil "	48·2% 48·2% nil "	34·1% 34·1% nil "

Non-pressure Type (2). The principle of all "V" notch meters is based on the original work of Professor James Thomson, of Queen's College, Belfast, and Glasgow University, who devised a formulae giving the relation between the height of the water flowing through a "V" notch and the weight of the water passing through the notch. The section of the flow is the same shape at all rates of flow, namely, a triangle, which simplifies the formulae. For a 90° triangular notch, Thomson's formulae can be expressed as—

$$\text{Cubic feet per min.} = 0{\cdot}305H^2 \times \sqrt{H}$$

where = the height in inches of the water in the notch.

The constant 0·305 was determined as the result of numerous experiments by Professor Thomson in 1860 and 1861. In 1907–1909, an investigation was carried out under the supervision of Professor Barr (of Glasgow University) to verify this constant; and the constant of 0·305 can be taken as accurate, at any rate, for heads ranging from 2 ins. to 7 ins. Thus, for a standard 90° "V" notch, the figures are as follows—

TABLE OF FLOWS THROUGH 90° V NOTCHES.

Depth in Notch in inches.	Flow per hour.		Depth in Notch in inches.	Flow per hour.	
	Galls.	Lbs.		Galls.	Lbs.
1 inch ..	114	1,140	9 inches ⅝	27,796	277,960
2 inches ..	648	6,480	10 ,, ⅝	36,174	361,740
3 ,, ..	1,783	17,830	11 ,, ⅝	45,903	459,030
4 ,, ..	3,661	36,610	12 ,, ⅝	56,872	568,720
5 ,, ..	6,394	63,940	13 ,, ⅝	69,471	694,710
6 ,, ..	10,086	100,860	14 ,, ⅝	83,611	836,110
7 ,, ..	14,829	148,290	15 ,, ⅝	99,351	993,510
8 ,, ..	20,706	207,060			

N.B.—1 gall. = 10 lbs.

In designing a water-meter based on these lines, the general principle is to erect a tank containing a "V" notch of known dimensions, through which the feed-water flows, and to devise a float mechanism which will give a continuous record of the varying level in the notch. From this record, the average height of the water in the "V" notch during the particular period is given, and from this the amount of water passing can easily be calculated from Thomson's formula.

LEA RECORDING AND INTEGRATING APPARATUS

This instrument, manufactured by Messrs. The Lea Recorder Co., Ltd., 28 Deansgate, Manchester, is an excellent example of an apparatus designed to measure the flow of water over V notches or rectangular weirs.

In its simplest form it consists of a notch, or measuring tank of suitable size, provided with notch partition in which is fixed the V-notch baffle partition, etc. Connected with the measuring compartment is a float chamber containing a copper float attached to which is a spindle which rises and falls with the level of water in the measuring compartment, in accordance with the amount of water flowing over the V notch.

This spindle has attached to it a rack which engages with a pinion fixed to a circular drum on the surface of which is cut a groove or screw-thread, corresponding to the "curve of flow" for the notch in use. This spiral drum serves to rectify the motion of the float so that a pen arm operated from a pin resting in the spiral groove, receives a motion which is directly proportional to the rate of flow over the notch. The pen traces a chart on a clock-driven drum and it is a simple matter to determine the total quantity which has been passed over any period, from the chart diagram, by means of a planimeter.

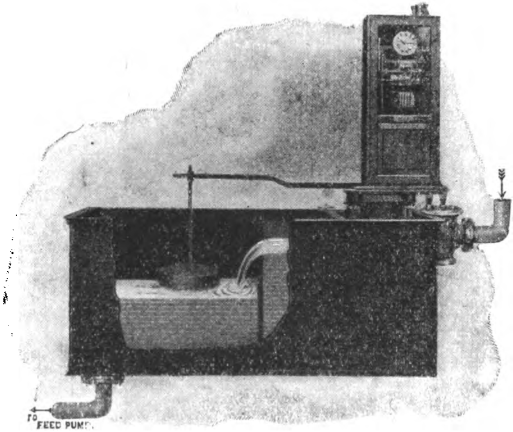


FIG. 38.—LEA V-NOTCH RECORDER.

The Lea Integrator consists of the instrument portions mentioned above, but, in addition, it is provided with a totalizing device consisting of a fine-toothed drum driven by a powerful clock movement. The teeth of this drum are undercut in the form of a spiral, proportionate to the amount of water passing through the notch. The counting box, or Integrator, is provided with a toothed wheel which engages with this spiral toothed drum, engaging more or less teeth as the flow increases or decreases.

These machines can be applied for measuring large flows of water in channels or streams by inserting a rectangular weir in the channel, and the grooved or indicating spiral drum in this case is constructed in accordance with the "curve of flow" for the rectangular weir used.

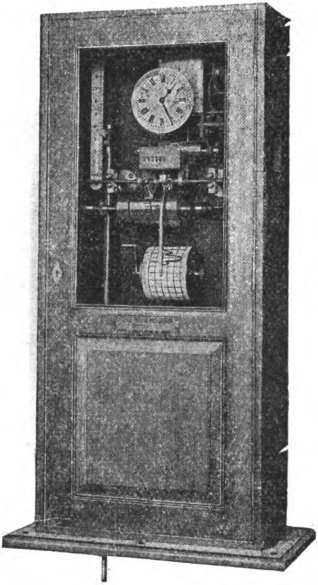


FIG. 39.—RECORDING MECHANISM OF LEA V-NOTCH APPARATUS.

PRESSURE TYPE 1.

In this type of meter, using a piston, the movement of a double-acting piston is indicated on a measuring device with a train of wheels, and each stroke of the piston corresponds to a definite volume of water. The movement of the piston is caused by the pressure of the boiler feed pump, the meter being placed between the boiler feed pump and the boiler plant, either before or after the economizers.

The following are particulars and illustrations of the "KENNEDY" METER.

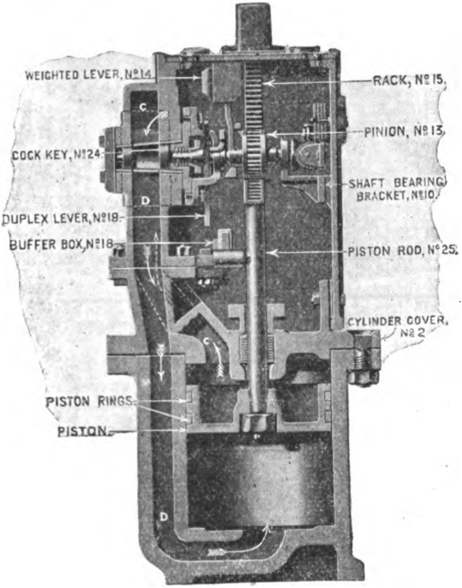


FIG. 40.—KENNEDY METER.

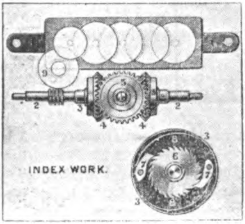


FIG. 41.—RECORDING MECHANISM OF KENNEDY METER.

DIMENSIONS OF NEW PATTERN HOT WATER METERS.

	½″	¾″	1″	1½″	2″
A	1′ 9″	2′ 3¾″	2′ 7⅜″	3′ 4¾″	3′ 11⅛″
B	1′ 3¾″	1′ 8⅝″	1′ 11″	2′ 5⅝″	2′ 10⅜″
C	10⅜″	1′ 0½″	1′ 0½″	1′ 2 1/16″	1′ 5 1/16″
D	1¾″	1⅞″	2⅛″	3″	4⅛″
E	11⅜″	1′ 1⅛″	1′ 2⅛″	1′ 4 13/16″	1′ 9 1/16″
F	10″	11½″	1′ 0⅜″	1′ 7¾″	1′ 10⅞″
G	4 5/16″	5¼″	5¼″	..	..
H	2½″	2¾″	2¾″	5½″ dia.	6½″ dia
Bolts	2—⅜″	2—½″	2—½″	4—⅝″	4—⅝″
C. to C. of Holes	2⅞″	3½″	3½″	4⅛″	5″
Dia. of Cyl	5⅞″	7⅜″	7⅜″	8⅞″	11⅞″
Weight	Cwts. Qrs. 1 2	Cwts. 2	Cwts. Qrs. Lbs. 2 2 14	Cwts. Qrs. 4 1	Cwts. Qrs. 7 1

In ½″, ¾″, and 1″ Meters the vertical line passes through the bolt holes.

The 1½″ and 2″ Meters have the vertical line between the bolt holes.

	3″	4″	6″	8″
A	1′ 4⅛″	1′ 5¼″	1′ 3½″	1′ 7½″
B	4′ 3¼″	5′ 2½″	5′ 10¼″	7′ 0¼″
C	1′ 8⅝″	2′ 1″	2′ 3″	2′ 7½″
D	7″	8″	11½″	1′ 1″
E	7″	8″	11½″	1′ 1″
F	2′ 0¾″	2′ 9″	2′ 11½″	3′ 9½″
G	8″ dia.	9″ dia.	12″ dia.	14½″ dia.
*H	2″	3″	..	..
Bolts	8—⅝″	8—⅝″	12—¾″	12—¾″
C. to C. of Holes	6½″	7½″	10¼″	12¾″
Dia. of Cyl.	14⅞″	17⅞″	20⅞″	24⅞″
Weight	Cwts. Qrs. 10 2	Cwts. 18	Cwts. Qrs. 26 2	Cwts. 45

In the 3″, 4″, 6″, and 8″ Meters the vertical line passes between the bolt holes.

The Flanges of Meters, 1½″ and upwards, are in accordance with British Standard, Table II, which now supersedes our former standard.

*Projection of Horns from back of Cock to prevent rolling during transit.

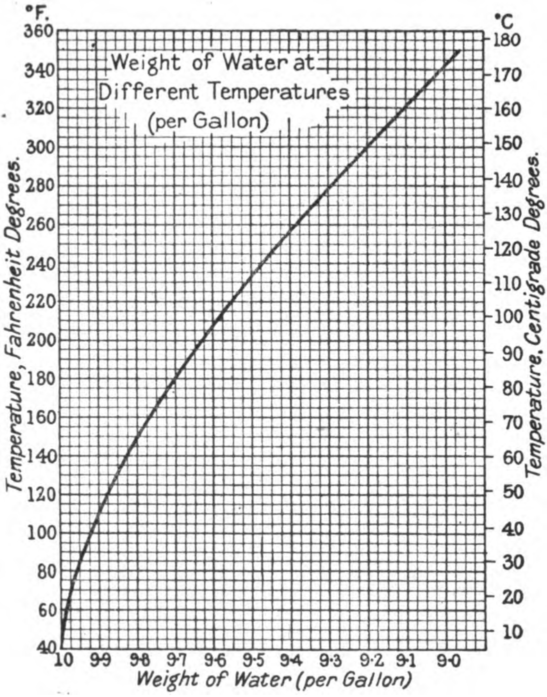


FIG. 42.—THE SPECIFIC GRAVITY OF WATER AT VARIOUS TEMPERATURES FOR USE WITH VOLUME METERS.

MEASUREMENT OF STEAM

STEAM METERS.

Steam meters are of great importance, not only from the point of view of generation of steam, but still more in connection with the utilization of steam. It is obviously of the greatest value to be able to keep a continuous record of the steam consumption of various departments of a factory, and also of various engines, turbines, special processes, etc.

It has, however, been a matter of the greatest difficulty to invent an accurate steam meter. The trouble is that the steam not only varies in pressure, but the amount is liable to fluctuate suddenly. Also the steam may contain water and, if superheated, the temperature of the superheat varies considerably. Within quite recent years, however, these difficulties have been largely overcome.

In measuring steam, it is necessary to use the "flow" principle, that is, to produce a pressure difference which varies in a definite and known relation to the rate of flow, and then to measure this small pressure difference in terms of the weight of steam.

For measuring this pressure difference, there are two fundamental principles, namely, "impact" and "change in velocity."

The "impact" method is the Pitot tube with its modifications, and the "change in velocity" is the Venturi tube and the orifice. The orifice and the Venturi tube are, however, on the same basic principle, when the orifice is used in a pipe and pressure connections made at the proper place.

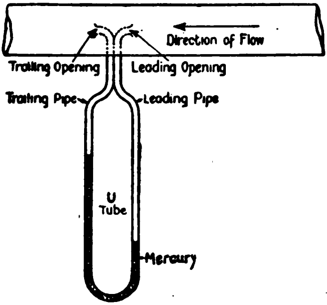


FIG. 43.—PITOT TUBE STEAM MEASUREMENT APPARATUS.

With regard to the small difference in pressure corresponding to the amount of steam passing, there are three principal methods used, namely, "V" tube, diaphragm, and liquid-sealed bell.

Very roughly, however, the action of a steam meter depends on the law governing the volume of a gas passing through an orifice in a plate when the difference in pressure on opposite sides of a plate is known. The flow of a gas varies as the square of the difference in pressure. An orifice plate is fitted between the two flanges of the pipe line, and a pipe from each side of the orifice plate leads to the meter. These pipes are kept full of condensed water, and communicate one to each side of a diaphragm "U" tube or sealed bell to record the pressure on opposite sides of the orifice plate. Of course, the weight of the steam passing through a pipe depends on the pressure, and this difference is compensated by a "Bourdon" tube which is connected up to the diaphragm rods by link-work to compensate the record.

The Bailey meter is constructed on the principle of measuring the drop in pressure across an orifice of "Monel" metal placed in the steam pipe. The difference in pressure, corresponding to the amount of steam passing, is recorded by means of a sealed bell floating in mercury, acting as a frictionless piston.

The British Thomson-Houston meter is on the "pitot"-tube principle, and depends on the fact that if the temperature and pressure of steam flowing through a pipe remain constant, then the quantity of steam passing is proportional to the velocity. A special shaped pipe is inserted in the steam pipe, as shown in the illustration, with one opening against the flow and the other opening with the flow, and connected by a "V" tube containing mercury.

The difference of pressure is measured by the difference in the height of the mercury, and corresponds to the amount of steam passing.

Elementary Form of Pitot Tube and Meter.

The Kent meter depends on the principle that the rate of flow of steam through a constriction placed in a steam main is approximately proportional to the square root of the difference of pressure across it, as with the "Bailey" meter. The recording part for the difference of pressure is a spring-controlled diaphragm.

The St. John meter depends on the principle that with a uniform difference of pressure on two sides of an orifice through which the steam is flowing and a constant initial pressure, the quantity of steam passing bears a direct relation to the size of the orifice. It is the only meter in which the whole of the steam passes direct through the meter mechanism.

The Sacro meter is also on the "Pitot"-tube principle, in which two ports in a special narrow disc inserted between the flanges of the steam pipe point in opposite directions in the flow of steam. The difference of pressure is recorded, and compensated for differences in boiler pressure by means of a swinging hollow cone partially filled with mercury and partially with water.

DETERMINATION OF MOISTURE IN STEAM

Steam Calorimeters.

It is not an easy matter to determine the amount of moisture in steam, although this question is obviously of the greatest importance.

Carpenter's Throttling Calorimeter.

This calorimeter consists essentially of a small vessel about 3 ins. diameter, to which the steam is supplied through a tapering or converging orifice, and which contains in the centre a deep thermometer socket to take the temperature of the steam. The instrument also has a mercury gauge for ascertaining the exact pressure. The steam passes through the calorimeter and is discharged from the bottom. When the pressure of high-pressure steam is lowered, some heat is liberated, which then evaporates any water in the steam and also raises the temperature of the steam above the equivalent temperature due to the pressure. If, therefore, there is only a very small amount of water in the steam the temperature of the steam when reduced in pressure is higher than it would normally be at the given reduced pressure. From this temperature the amount of moisture in the steam can be worked out with a series of curves. This calorimeter can only be used when the gauge pressure is over 35 lbs. and less than about 285 lbs., and when the amount of moisture in the steam is between about 2·7 and 7·5 per cent.

CARPENTER'S SEPARATING CALORIMETER.

This instrument depends on the principle of mechanically separating the moisture in the steam. It consists of two vessels, one inside the other, the outer one being a steam jacket. The steam passes into the inner vessel through a valve in the top. The water in the steam is thrown downwards into a meshed cup, and the course of the steam is then suddenly changed through an angle of nearly 180°, which causes the water in the steam—because it is much heavier—to be thrown through the meshes of the cup into a space below the inner chamber. The steam then passes upwards, enters the top of the outer chamber, and is discharged through the bottom. The pressure in the outer chamber and the flow of steam in a given time is shown by two scales on the attached pressure gauge. The scale for showing the flow of steam is a graduated water gauge, which gives directly the weight of steam in ten minutes' flow.

4. PYROMETERS

The simplest form is the ordinary mercurial thermometer. For measuring superheat temperatures, the thermometer must be made of specially tough glass, and the space above the mercury filled with nitrogen under high pressure. This allows the thermometer to be used for high temperatures, and tends to prevent the mercury thread "breaking."

Recording mercurial thermometers are also available, in which a continuous record of the temperature is recorded on a chart.

"Bristol" recording pyrometers depend on the principle of exposing to the heat source a sensitive bulb containing the saturated vapour of a liquid. The pressure in the bulb rises or falls with the temperature, and the bulb is connected to the recording instrument by a small flexible copper pipe 25 ft. long —or more. The change in the pressure corresponding to the change in temperature actuates the pen lever of the recording apparatus and records the temperature direct on a chart in the usual manner.

A very efficient type of pyrometer for practically any temperature, and for portable or permanent use, is the thermoelectric. The general principle of these instruments is that a junction of two dissimilar metallic wires, when heated, causes a small current to flow when the circuit is completed. The amount of the current is proportional to the heat, and a delicate galvanometer is inserted in the circuit, which is graduated to read direct in ° F. instead of, say, milli-amps. The wires are contained in a porcelain or quartz tube enclosed in a steel tube forming a pointer with two terminals. These are coupled up by standard coils of wire to the galvanometer, and the temperature read off. The composition of the dissimilar metals used depends on the temperature required. The most expensive installation for temperatures up to, say, about 2,500° F. is platinum and an alloy of platinum-iridium. Such a connection is indestructible, and the wires at all times have a high market value.

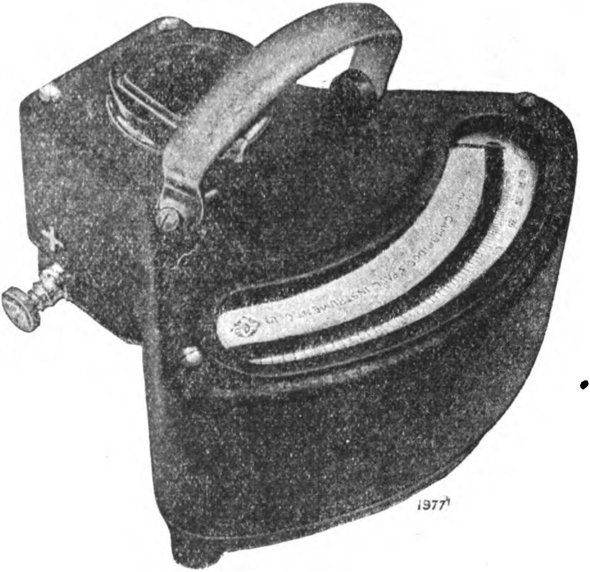


FIG. 44.—PYROMETER.

For temperatures up to, say, 1500° F., less expensive metals can be used, such as copper, iron, "constantan" alloy, and "Titan" alloy.

The illustration shows an ordinary indicating instrument for portable use, and a complete modern recording instrument,

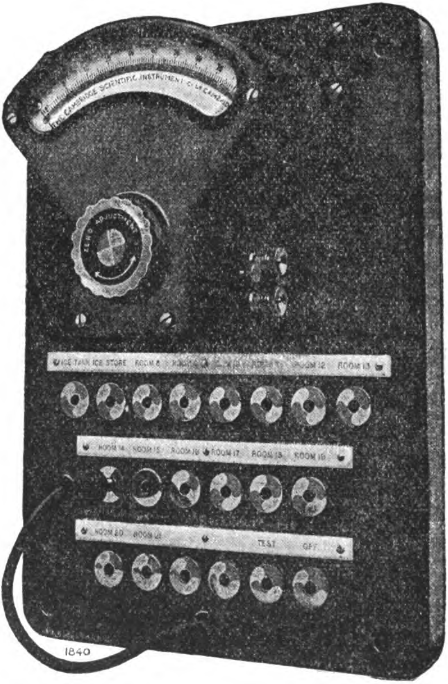

CAMBRIDGE SCIENTIFIC INSTRUMENT Co Ltd CAMBRIDGE
ZERO ADJUSTMENT
ICE TANK ICE STORE ROOM 6
ROOM 12 ROOM 13
ROOM 14 ROOM 15 ROOM 16 ROOM 17 ROOM 18 ROOM 19
ROOM 20 ROOM 21
TEST OFF
1840

FIG. 45.—PYROMETER.

a double-thread recorder with two independent galvanometers side by side on a double width chart with a two-colour thread mechanism by the Cambridge & Paul Scientific Instrument Co., Cambridge.

For permanent installation on a large boiler plant, the "distance electrical pyrometers" are to be strongly recommended. The pointers are placed permanently with the various positions necessary, such as feed-water before and after each economizer, flue gas before and after each economizer and chimney base, saturated and superheated steam mains, and so on. The pointers are all wired to a central indicating instrument, so that by inserting a plug the whole temperature range, even for 30 or 40 "points," can be read off in succession.

The principle of these instruments is that the resistance of a platinum wire to the passage of an electric current varies with the temperature of the wire according to a well-known law. A current is passed continually through the circuit from a small 4-volt accumulator on the "Wheatstone Bridge" principle, and the pointer, on being heated, alters the resistance proportional to the temperature, which is indicated on a sensitive galvanometer.

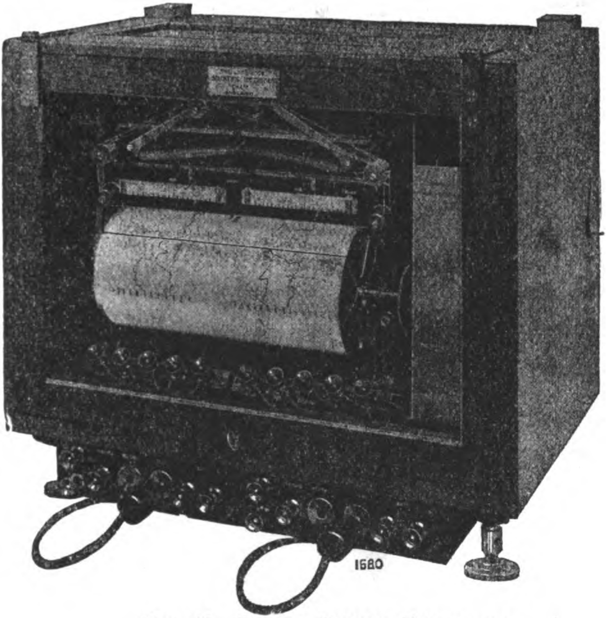


FIG. 46.—DOUBLE-THREAD PYROMETER.

5. COAL-WEIGHING APPLIANCES (AUTOMATIC)

These machines can be fitted on to each individual boiler or to the coal-conveying plant for a number of boilers, and are so arranged that all the coal supplied passes through them.

The Avery machine is supplied in various types, either a stationary type—to be fitted to each boiler—or a travelling machine for feeding a number of boilers. The coal is automatically fed into the hopper and, when 2 cwts. has been received, the machine automatically releases its base and allows the 2 cwts. of coal to fall through and down into the mechanical stoker hopper or elsewhere as required. The machine is fitted with a counter, which records the number of times the machine releases and, therefore, the weight of the coal. The machine is worked entirely by gravity, and requires no power to drive it. Also it cannot be tampered with, and is provided with a locking cam, which prevents coal running through unweighed.

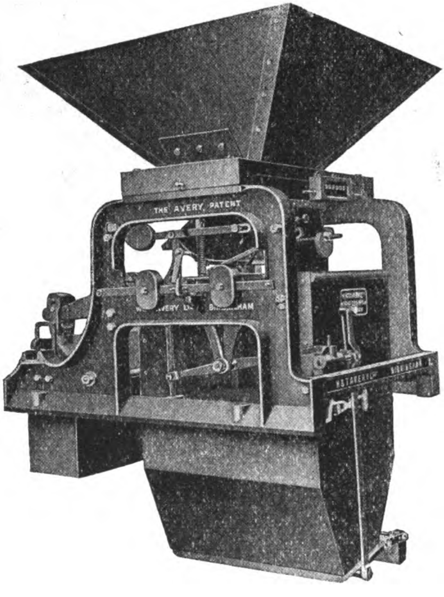
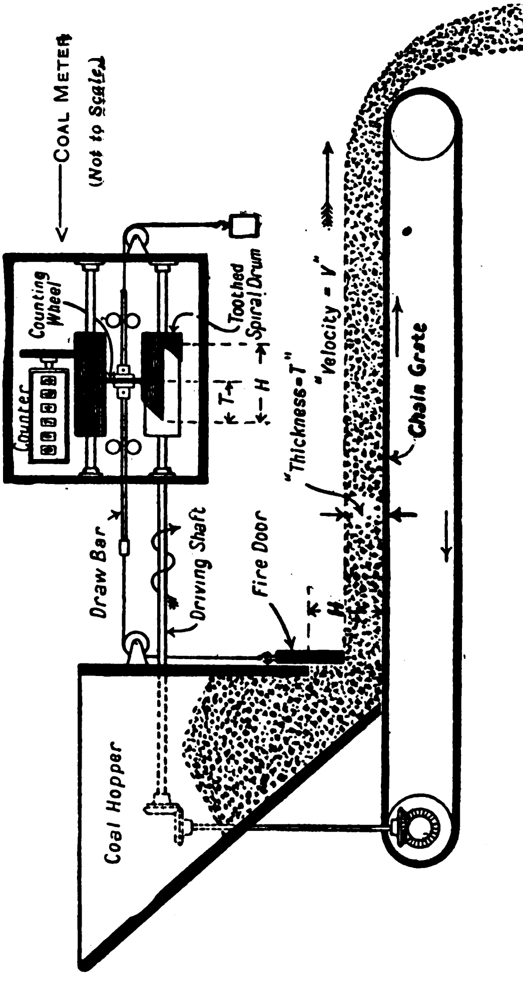


FIG. 47.—AVERY COAL WEIGHING MACHINE.

THE "LEA" COAL METER—

Is a special machine for chain grate stokers only. Its action is based on the theory that when coal is supplied to a boiler by means of a chain-grate stoker, the amount



(*N.B. The actual size of the meter is* 24" × 12" × 6".)

FIG. 48.—LEA COAL METER. DIAGRAM SHOWING PRINCIPLES OF OPERATION.

of fuel passing under the firedoor may be regarded as a stream with a constant width, but the depth and velocity of the stream are subject to variation. The width of the stream is the width of the chain-grate, the depth is the thickness of the fire, and the speed is the rate of travel of the chain-grate.

If W = the width of the stream in feet;
T = the thickness or depth in feet;
V = the velocity of the stream in feet per hour;
Then the cubic feet per hour

$$= W \times T \times V = \text{cross-sectional area} \times V.$$

As the width W is constant, and T and V are the only variables, all that is required is some form of automatic integrating mechanism which will at all times take into account the two items T and V. This is what the inventors claim that the Lea recorder actually does. As will be seen from the diagram, the motion of the grate is transmitted by gearing to a spirally-toothed drum, the pitch of the teeth of which is equal, or proportionate to, the maximum lift of the firedoor, or the thickness of the fire H. A toothed counting wheel gearing with the spiral drum below, and with a counting box above is mounted on a rod, or drawbar, directly connected with the firedoor, and as the latter is opened or closed, the counting wheel is moved to and fro laterally across the spiral drum, which causes the wheel to revolve, more or less, according to its lateral position.

The actual size of the meter is 24″ × 12″ × 6″. It is usually fixed on the front of the hopper or on the worm wheel casing of the stoker.

6. THE MEASUREMENT OF CO_2

CO_2 RECORDERS.

An effective and necessary check on efficient combustion is obtained from an analysis of the flue gases to determine the percentage of carbon dioxide (CO_2). The theory of this is as follows—

If coal consisted of pure carbon, and it were completely burned with the theoretical amount of air, 1 lb. of carbon would combine with the oxygen to give 14,600 thermal units of heat, and the result would be 3·66 lbs. of CO_2. The analysis of the air (the flue gases) after combustion would show 21 per cent. CO_2 and 79 per cent. nitrogen, since the air originally contains 21 per cent. oxygen and 79 per cent. nitrogen, and the oxygen would be totally used up in burning the carbon under perfect conditions. If excess air were used, the percentage of CO_2 would be less than 21 per cent., since there would be a large amount of oxygen still in the flue gases. Also, if not enough air was used, part of the carbon would be burned only to carbon monoxide (CO), giving only 4,400 B.Th.U. per lb. of carbon, and again the percentage of CO_2 would be less than 21 per cent., since part of the flue gases would be CO. Hence, by analysing the flue gases for CO_2, the efficiency of the combustion can be determined. In practice, coal is, of course, not pure carbon, and contains smaller quantities of other combustible substances, such as sulphur and hydrogen, besides ash and water. Anthracite (say, 92½–95 per cent. carbon) or coke (say, 90·0 per cent. carbon), containing a high percentage of carbon, will give, under ideal conditions, as high as 19½ per cent. CO_2 in the flue gases. Ordinary steam coal contains, say, 80–90 per cent. carbon and considerable amounts of hydrogen, say, 4–5 per cent., which burns to water (H_2O), consequently the highest percentage of CO_2 possible is, say, 12–16½ per cent.

A Combustion Recorder, that is a machine for continuously and automatically taking a sample of flue gas, analysing it and recording the result on a chart, say, twenty times an hour, is consequently an essential instrument for the boiler plant. The results for CO_2 on most boiler plants are very poor, the average probably not being higher than 6–7 per cent.

The causes of low percentage of CO_2 are due to—

(1) *Excess air*, caused by poor firing, such as hollow and uneven fires allowing a large excess of air to pass through. Also by leaky and defective boiler settings, so that, although the combustion may be good, the percentage of CO_2 in the flue gases is immediately reduced by air leakage. Also by bad regulation of the draught, so that excess air is pulled bodily through the whole fire. This causes a serious loss of efficiency, because of the cooling effect of this excess air, which serves no useful purpose.

(2) *Lack of air*, so that part of the coal is burnt to CO, and black smoke is formed. This can be caused by lack of draught, or by poor firing, in which the fires are so thick that the air cannot get through. The loss in efficiency by the formation of CO is exceptionally serious, as will be seen by the heat of formation of CO_2 (14,600 units) as compared with CO (4,400 u.).

Both these conditions, that is excess air and formation of CO, can exist side by side in the same fire. Also a flue gas can have a very high percentage of CO_2, say, 15–16 per cent., and CO at the same time. For this reason, such a plant may be less efficient than a similar plant with only, say, 7–8 per cent. CO_2.

What is necessary is high percentage of CO_2 with no CO, and high CO_2 alone is not necessarily efficient. For this reason, the CO_2 recorder must be used with discretion. The most economical conditions are, roughly, 40 per cent. excess air, 13 per cent. CO_2, and less than 0·2 per cent. CO.

An approximate idea of the losses due to low CO_2 is given by the following curve, based on normal temperature in the flue gases and absence of CO.

Thus, for example, with 7 per cent. CO_2, the loss is 25 per cent. of the coal bill; and if this is increased to 12 per cent., the loss is only 15 per cent., giving a saving of 10 per cent. in the coal bill.

The principle on which most CO_2 recorders work is to pass a measured volume (100 cc.) of the flue gas through some chemical absorbent, to absorb the CO_2, and then to re-measure the residual gas, the loss in volume giving the percentage of CO_2.

The usual machines work, roughly, by means of a dribble of water, which enters a receiver until the latter is full and then suddenly empties itself by starting a syphon, which causes a suction bell to fall and fill itself with flue gas. The bell then slowly rises and bubbles the flue gas through a solution of caustic potash, and then passes into another bell which rises. The height to which the latter rises is short of a datum mark, depending on how much CO_2 has been absorbed, and the difference is the percentage of CO_2. This rising measuring bell works a pen mechanism, which records the result on a chart driven by a clock in the usual manner.

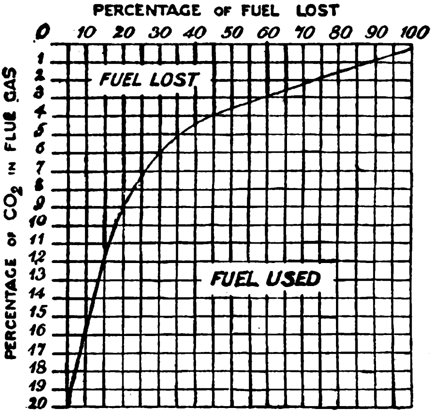


FIG. 49.—CURVE OF LOSS DUE TO CO_2.

The "Mono" machine, however, works with mercury instead of water as a driving agent, but otherwise is on the same principle.

A new CO_2 recorder on a novel plan has just been put on the market by *Messrs. The Cambridge Scientific Instrument Co., Ltd.*, being an *Electrical CO_2 Recorder*, the invention of Dr. G. A. Shakespear. The principle consists in passing a current through two platinum wires, one in air and one in flue gas. The hot wires lose heat, and the rate of cooling depends on the density of the surrounding atmosphere, the flue gases containing CO_2 being different in this respect from air. This causes a difference in the current flowing, which is indicated on a galvanometer so graduated as to read the percentage of CO_2.

In general, the CO_2 recorder must be placed in the fire-hole in full view of the fireman, for whose benefit the machine is intended. If it is out of the fire-hole, the firemen will not use it.

It must be connected up with suitable piping to each boiler, as near the fires as possible. In a " Lancs " or similar type of boiler, the side flue or downtake is the best position, and in a tubular boiler at the back of the fires. Each pipe must be fitted with a valve, so that any boiler can be put on at will. The pipe circuit can include connections to the main flue, the chimney base, before and after the economizers, or any other positions. Each make of machine is provided with some kind of filtering apparatus, to take out the coal and ash dirt from the flue gases before analysing. Also, for most machines, the whole flue gas pipe circuit is connected to the chimney base so that there is a continuous travel of gas from the furnaces to the chimney. The machine, therefore, draws a fresh charge of flue gas at each stroke, and there is no " lag " in the pipe circuit. Most machines will analyse at a rate of up to twenty analyses per hour, and the speed can be regulated as desired.

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